DESIGN OF FLUID SYSTEMS

HOOK-UPS

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Spirax Sarco is the recognized industry standard for knowledge and products and for over 85 years has been committed to servicing the steam users worldwide. The existing and potential applications for steam, water and air are virtually unlimited. Beginning with steam generation, through distribution and utilization and ultimately returning condensate to the boiler, Spirax Sarco has the solutions to optimize steam system performance and increase productivity to save valuable time and money.

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Spirax Sarco draws upon its worldwide resources of over 3500 people to bring complete and thorough service to steam users. This service is built into our products as a performance guarantee. From initial consultation to effective solutions, our goal is to manufacture safe, reliable products that improve productivity. With a quick, responsive team of sales engineers and a dedicated network of local authorized distributors Spirax Sarco provides quality service and support with fast, efficient delivery.

Reliable steam system components are at the heart of Spirax Sarco's commitment. Controls and regulators for ideal temperature, pressure and flow control; steam traps for efficient drainage of condensate for maximum heat transfer; flowmeters for precise measurement of liquids; liquid drain traps for automatic and continuous drain trap operation to boost system efficiency; rotary filters for increased productivity through proper filtering of fluids; condensate recovery pumps for effective condensate management to save water and sewage costs; stainless steel specialty products for maintaining quality and purity of steam; and a full range of pipeline auxiliaries, all work together to produce a productive steam system. Spirax Sarco's new line of engineered equipment reduces installation costs with prefabricated assemblies and fabricated modules for system integrity and turnkey advantages.

From large oil refineries and chemical plants to local laundries, from horticulture to shipping, for hospitals, universities, offices and hotels, in business and government, wherever steam, hot water and compressed air is generated and handled effectively and efficiently, Spirax Sarco is there with knowledge and experience.

For assistance with the installation or operation of any Spirax Sarco product or application, call toll free:

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How to Use This Book

Selection of the most appropriate type and size of control valves, steam traps and other fluid control valves, steam traps and other fluid control equipment, and installation in a hook up enabling these components of a system to operate in an optimal manner, all bear directly on the efficiency and economy obtainable in any plant or system.

To help make the best choice, we have assembled into this book the accumulation of over 85 years of experience with energy services in industrial and commercial use. The hook ups illustrated have all been proven in practice, and the reference information included is that which we use ourselves when assisting customers choose and use our products.

The Case in Action stories dispersed throughout this book are actual applications put to the test by steam users throughout the country. Their stories are testimonials to the products and services Spirax Sarco offers and the benefits they have received from utilizing our knowledge and services.

The Hook Up Book is divided into three sections:

Section I is a compilation of engineering data and information to assist in estimating loads and flow rates, the basic parameters which enable the best choice when selecting sizes.

Section II illustrates how the services and control equipment can be assembled into hook ups to best meet the particular needs of each application.

Section III is a summary of the range of Spirax Sarco equipment utilized in the hook ups. Although it is not a complete catalog of the entire range, it does describe generically the capabilities and limitations which must be remembered when making proper product choices.

Most application problems will be approached in the same order. Section I will enable the load information to be collected and the calculations made so that sizing can be carried out; Section II will make sure that the essentials of the hook up, or combination of hook ups, are not overlooked; and Section III will serve as a guide to the complete equipment catalog so that the most suitable equipment can readily be selected.

The Hook Up Book is intended to serve as a reference for those actively engaged in the design, operation and maintenance of steam, air and liquid systems. It is also intended as a learning tool to teach engineers how to design productive steam systems, efficiently and cost effectively.

We gratefully acknowledge the valuable contributions made by our field engineers, representatives, application engineers, and customers to the body of accumulated experience contained in this text.

Table of Contents

SYSTEM DESIGN INFORMATION

Section 1

The Working Pressure in the Boiler and the Mains

Steam should be generated at a pressure as close as possible to that at which the boiler is designed to run, even if this is higher than is needed in the plant. The reasoning behind this is clear when consideration is given to what happens in the water and steam space within the boiler. Energy flows into the boiler water through the outer surface of the tubes, and if the water is already at saturation temperature, bubbles of steam are produced. These bubbles then rise to the surface and break, to release steam into the steam space.

The volume of a given weight of steam contained in the bubbles depends directly on the pressure at which the boiler is operating. If this pressure is lower than the design pressure, the volume in the bubbles is greater. It follows that as this volume increases, the apparent water level is raised. The volume of the steam space above the water level is thereby reduced. There is increased turbulence as the greater volume of bubbles break the surface, and less room for separation of water droplets above the surface. Further, the steam moving towards the crown or steam take-off valve must move at greater velocity with a higher volume moving across a smaller space. All these factors tend to encourage carryover of water droplets with the steam.

There is much to be said in favor of carrying the steam close to the points of use at a high pressure, near to that of the boiler. The use of such pressure means that the size of the distribution mains is reduced. The smaller mains have smaller heat losses, and better quality steam at the steam users is likely to result.

Pressure reduction to the values needed by the steam using equipment can then take place through pressure reducing stations close to the steam users themselves. The individual reducing valves will be smaller in size, will tend to give tighter control of reduced pressures, and emit less noise. Problems of having a whole plant dependent on a single reducing station are avoided, and the effects on the steam users of pressure drops through the pipework, which change with varying loads, disappear.

Table 1: Steam Pipe Sizing for Steam Velocity

Capacity of Sch. 80 Pipe in lb/hr steam

Pressure psi	Velocity ft/sec	1/2"	3/4"	1"	1 1/4"	1 1/2"	2"	2 1/2"	3"	4"	5"	6"	8"	10"	12"
5	50	12	26	45	70	100	190	280	410	760	1250	1770	3100	5000	7100
	80	19	45	75	115	170	300	490	710	1250	1800	2700	5200	7600	11000
	120	29	60	110	175	245	460	700	1000	1800	2900	4000	7500	12000	16500
10	50	15	35	55	88	130	240	365	550	950	1500	2200	3770	6160	8500
	80	24	52	95	150	210	380	600	900	1500	2400	3300	5900	9700	13000
	120	35	72	135	210	330	590	850	1250	2200	3400	4800	9000	14400	20500
20	50	21	47	82	123	185	320	520	740	1340	1980	2900	5300	8000	11500
	80	32	70	120	190	260	520	810	1100	1900	3100	4500	8400	13200	18300
	120	50	105	190	300	440	840	1250	1720	3100	4850	6750	13000	19800	28000
30	50	26	56	100	160	230	420	650	950	1650	2600	3650	6500	10500	14500
	80	42	94	155	250	360	655	950	1460	2700	3900	5600	10700	16500	23500
	120	62	130	240	370	570	990	1550	2100	3950	6100	8700	16000	25000	35000
40	50	32	75	120	190	260	505	790	1100	1900	3100	4200	8200	12800	18000
	80	51	110	195	300	445	840	1250	1800	3120	4900	6800	13400	20300	28300
	120	75	160	290	460	660	1100	1900	2700	4700	7500	11000	19400	30500	42500
60	50	43	95	160	250	360	650	1000	1470	2700	3900	5700	10700	16500	24000
	80	65	140	250	400	600	1000	1650	2400	4400	6500	9400	17500	27200	38500
	120	102	240	410	610	950	1660	2600	3800	6500	10300	14700	26400	41000	58000
80	50	53	120	215	315	460	870	1300	1900	3200	5200	7000	13700	21200	29500
	80	85	190	320	500	730	1300	2100	3000	5000	8400	12200	21000	33800	47500
	120	130	290	500	750	1100	1900	3000	4200	7800	12000	17500	30600	51600	71700
100	50	63	130	240	360	570	980	1550	2100	4000	6100	8800	16300	26500	35500
	80	102	240	400	610	950	1660	2550	3700	6400	10200	14600	26000	41000	57300
	120	150	350	600	900	1370	2400	3700	5000	9100	15000	21600	38000	61500	86300
120	50	74	160	290	440	660	1100	1850	2600	4600	7000	10500	18600	29200	41000
	80	120	270	450	710	1030	1800	2800	4150	7200	11600	16500	29200	48000	73800
	120	175	400	680	1060	1520	2850	4300	6500	10700	17500	26000	44300	70200	97700
150	50	90	208	340	550	820	1380	2230	3220	5500	8800	12900	22000	35600	50000
	80	145	320	570	900	1250	2200	3400	4900	8500	14000	20000	35500	57500	79800
	120	215	450	850	1280	1890	3400	5300	7500	13400	20600	30000	55500	85500	120000
200	50	110	265	450	680	1020	1780	2800	4120	7100	11500	16300	28500	45300	64000
	80	180	410	700	1100	1560	2910	4400	6600	11000	18000	26600	46000	72300	100000
	120	250	600	1100	1630	2400	4350	6800	9400	16900	25900	37000	70600	109000	152000

The appropriate size of pipe to carry the required amount of steam at the local pressure must be chosen, since an undersized pipe means high pressure drops and velocities, noise and erosion, while a generously sized pipe is unnecessarily expensive to install and heat losses from it will also be greater than they need be.

Steam pipes may be sized either so that the pressure drop along them is below an acceptable limit, or so that velocities along them are not too high. It is convenient and quick to size short mains and branches on velocity, but longer runs of pipe should also be checked to see that pressure drops are not too high.

Steam Line Velocities

In saturated steam lines, reasonable maximum for velocities are often taken at 80/120 ft. per second or 4800/7200 fpm. In the past, many process plants have used higher velocities up to 200 ft. per second or 12,000 fpm, on the basis that the increased pipe noise is not a problem within a process plant. This ignores the other problems which accompany high velocities, and especially the erosion of the pipework and fittings by water droplets moving at high speed. Only where appreciable superheat is present, with the pipes carrying only a dry gas, should the velocities mentioned be exceeded. Velocity of saturated steam in any pipe may be obtained from either Table 1, Fig. 1 or calculated in ft. per minute using the formula:

Formula For Velocity Of Steam In Pipes

$$V = \frac{2.4Q\ Vs}{A}$$

Where:

- V - Velocity in feet per minute
- Q - Flow lbs./hr. steam
- Vs - Sp. Vol. in cu. ft./lb. at the flowing pressure
- A - Internal area of the pipe—sq. in.

Steam Piping For PRV's and Flash Vents

Velocity in piping other than steam distribution lines must be correctly chosen, including pressure reducing valve and flash steam vent applications.

A look at Steam Properties (Table 3) illustrates how the specific volume of steam increases as pressure is reduced. To keep reducing valve high and low pressure pipe velocity constant, the downstream piping cross-sectional area must be larger by the same ratio as the change in volume. When downstream pipe size is not increased, low pressure steam velocity increases proportionally. For best PRV operation, without excessive noise, long straight pipe runs must be provided on both sides, with piping reduced to the valve then expanded downstream gradually to limit approach and exit steam velocities to 4000/ 6000 fpm. A sizing example is given in Fig. 1.

Line velocity is also important in discharge piping from steam traps where two-phase steam/condensate mixtures must be slowed to allow some gravity separation and reduce carryover of condensate from flash vent lines. Here line velocities of the flash steam should not exceed 50/66 ft. per second. A much lower velocity must be provided for separation inside the flash vessel by expanding its size. The flash load is the total released by hot condensate from all traps draining into the receiver. For condensate line sizing example, see page 46 and see page 43 for vent line sizing example.

Sizing Steam Lines On Velocity

Fig. 1 lists steam capacities of pipes under various pressure and velocity conditions.

EXAMPLE: Given a steam heating system with a 100 psig inlet pressure ahead of the pressure reducing valve and a capacity of 1,000 pounds of steam per hour at 25 psig, find the smallest sizes of upstream and downstream piping for reasonable quiet steam velocities.

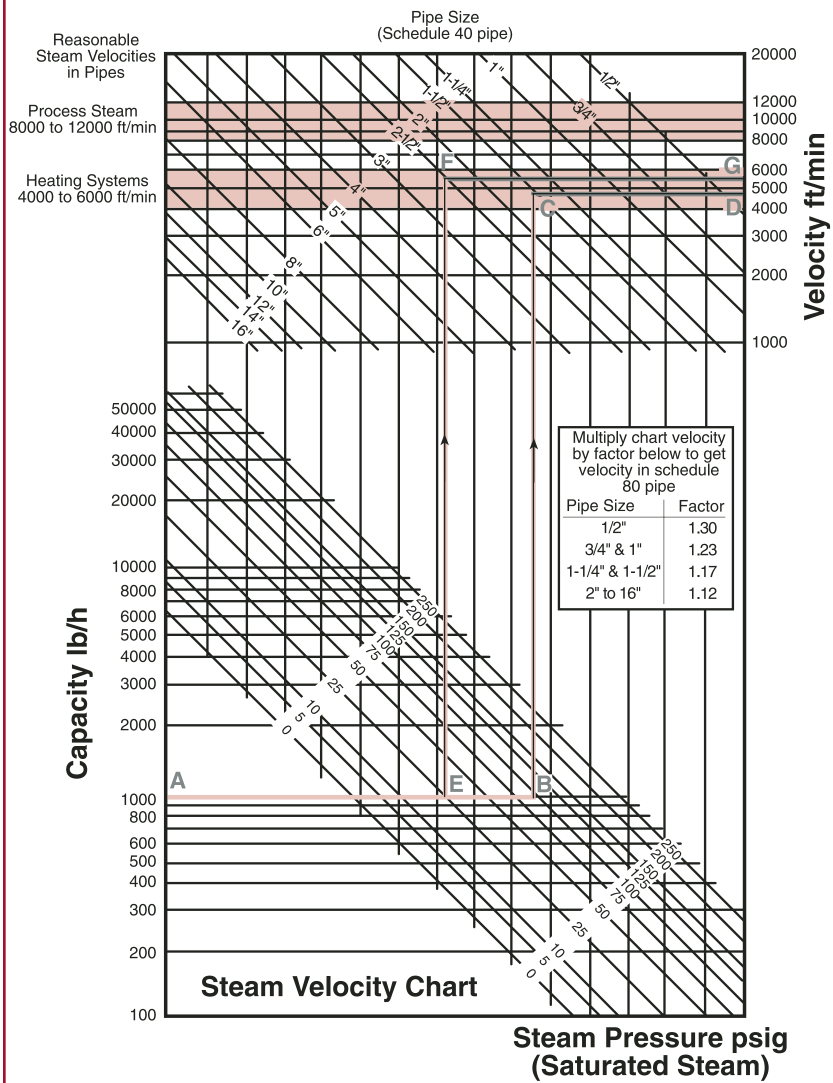
Upstream Piping Sizing

Enter the velocity chart at A for 1,000 pounds per hour. Go over to point B where the 100 psig diagonal line intersects. Follow up vertically to C where an intersection with a diagonal line falls inside the 4,000-6,000 foot-per-minute velocity band. Actual velocity at D is about 4,800 feet per minute for 1-1/2 inch upstream piping.

Downstream Piping Sizing

Enter the velocity chart at A for 1,000 pounds per hour. Go over to point E where the 25 psig diagonal line intersects. Follow up vertically to F where an intersection with a diagonal line falls inside the 4,000-6,000 foot-per-minute velocity band. Actual velocity at G is 5,500 feet per minute for 2-1/2 inch downstream piping.

Figure 1: Steam Velocity Chart



Pressure Drop in Steam Lines

Always check that pressure drop is within allowable limits before selecting pipe size in long steam mains and whenever it is critical. Fig. 2 and Fig. 3 provide drops in Sch. 40 and Sch. 80 pipe. Use of the charts is illustrated in the two examples.

EXAMPLE 1

What will be the smallest schedule 40 pipe that can be used if drop per 100 feet shall not exceed 3 psi when flow rate is 10,000 pounds per hour, and steam pressure is 60 psig?

Solution:

1. Find factor for steam pressure in main, in this case 60 psig. Factor from chart = 1.5.
2. Divide allowable pressure drop by factor 3÷1.5 = 2 psi.
3. Enter pressure drop chart at 2 psi and proceed horizontally to flow rate of 10,000 pounds per hour. Select pipe size on or to the right of this point. In this case a 4" main.

EXAMPLE 2

What will be the pressure drop per 100 feet in an 8" schedule 40 steam main when flow is 20,000 pounds per hour, and steam pressure is 15 psig?

Solution:

Enter schedule 40 chart at 20,000 pounds per hour, proceed vertically upward to 8" pipe curve, then horizontally to pressure drop scale, read 0.23 psi per 100 feet. This would be the drop if the steam pressure were 100 psig. Since pressure is 15 psig, a correction factor must be used.

Correction factor for 15 psig = 3.6
0.23 x 3.6 = 0.828 psi drop per 100 feet for 15 psig

Figure 2: Pressure Drop in Schedule 40 Pipe

100 psig Saturated Steam

For other pressures use correction factors

psi	0	2	5	10	15	20	30	40	60	75	90	100	110	125	150	175	200	225	250	300	350	400	500	600
factor	6.9	6.0	5.2	4.3	3.6	3.1	2.4	2.0	1.5	1.3	1.1	1.0	0.92	0.83	0.70	0.62	0.55	0.49	0.45	0.38	0.33	0.29	0.23	0.19

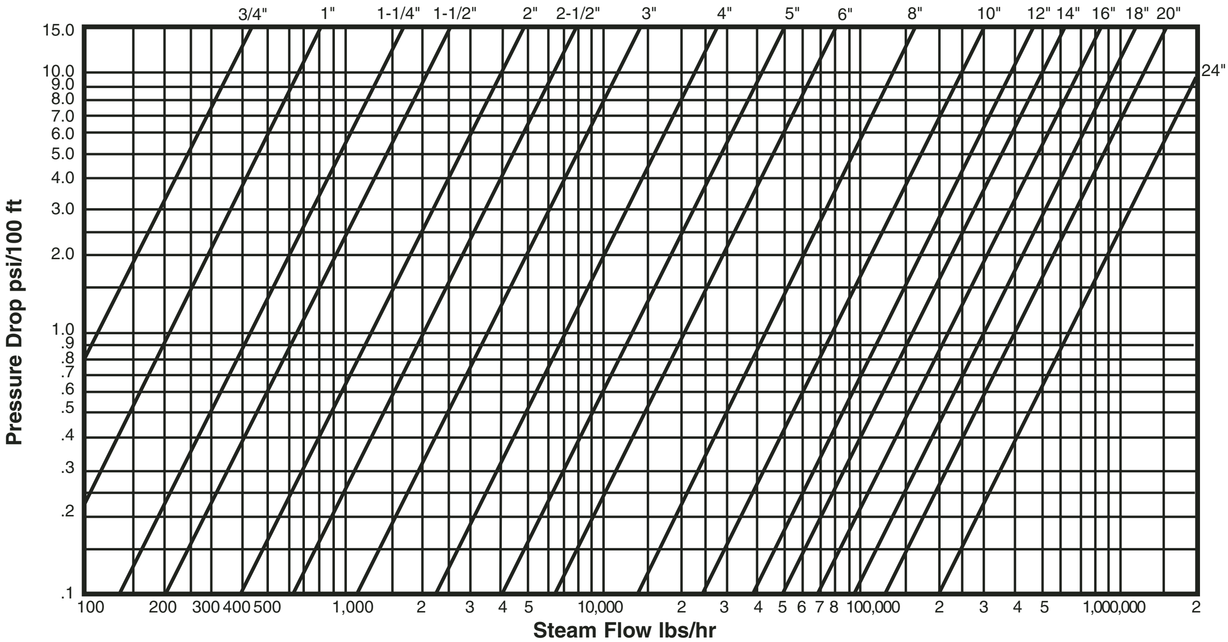
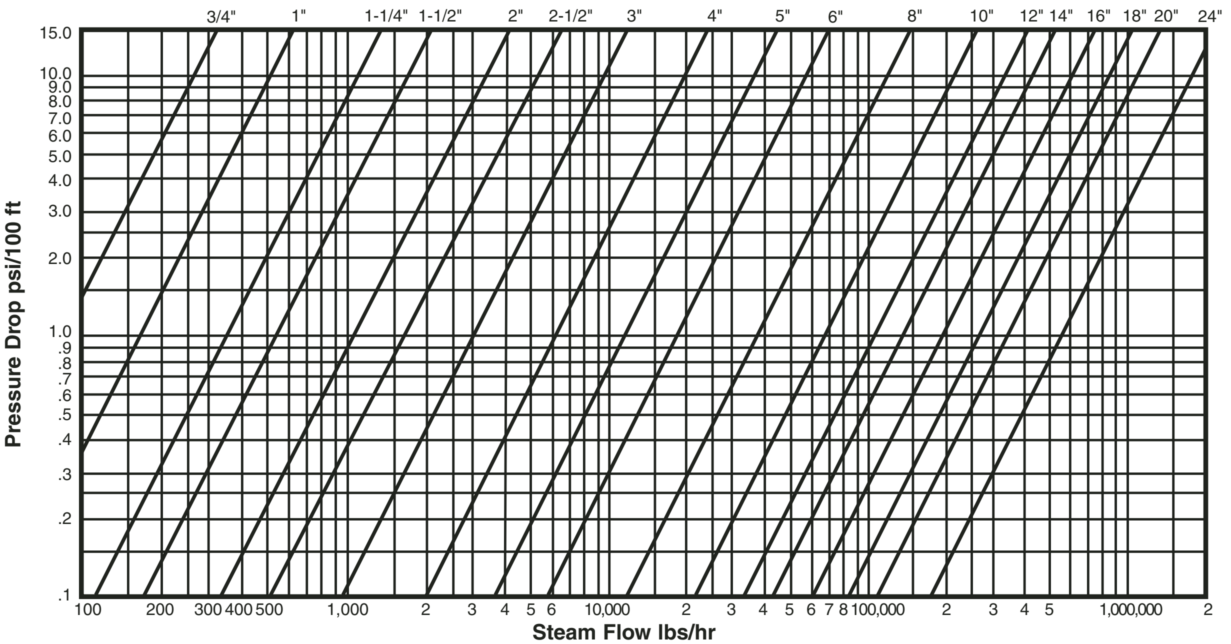


Figure 3: Pressure Drop in Schedule 80 Pipe

100 psig Saturated Steam

For other pressures use correction factors

psi	0	2	5	10	15	20	30	40	60	75	90	100	110	125	150	175	200	225	250	300	350	400	500	600
factor	6.9	6.0	5.2	4.3	3.6	3.1	2.4	2.0	1.5	1.3	1.1	1.0	0.92	0.83	0.70	0.62	0.55	0.49	0.45	0.38	0.33	0.29	0.23	0.19



Sizing Superheated Mains

When sizing steam mains for superheated service, the following procedure should be used. Divide the required flow rate by the factor in Table 2. This will give an equivalent saturated steam flow. Enter Fig. 1, Steam Velocity Chart on page 4 to select appropriate pipe size. If unable, then use the formula on page 3 to calculate cross sectional area of the pipe and then Tables 38 and 39, page 81, to select the pipe size which closely matches calculated internal transverse area.

Example:

Size a steam main to carry 34,000 lb/h of 300 psig steam at a temperature of 500° F.

From Table 2 the correction factor is .96. The equivalent capacity is

$$\frac{34{,}000}{.96} = 35{,}417 \text{ lb/h.}$$

Since 300 psig is not found on Fig. 1, the pipe size will have to be calculated. From the formula on page 3:

$$V = \frac{2.4 \times Q \times Vs}{A}$$

Solving for area the formula becomes:

$$A = \frac{2.4 \times Q \times Vs}{V}$$

Select a velocity of 10,000 ft/min. (which is within the process velocity range of 8,000 - 12,000 ft/min.) and determine Vs (specific volume) of 1.47 ft^3/lb (from the Steam Table on page 7). The formula is now:

$$A = \frac{2.4 \times 35{,}417 \times 1.47}{10{,}000} = 12.5 \text{ in}^2$$

From Tables 38 and 39 (page 81) the pipe closest to this area is 4" schedule 40 or 5" schedule 80.

Table 2: Superheated Steam Correction Factor

Gauge Pressure PSI	Saturated Temp. ˚F	340	360	380	400	420	440	460	480	500	520	540	560	580	600	620	640	660	680	700	720	740	760
		TOTAL STEAM TEMPERATURE IN DEGREES FARENHEIT																					
15	250	.99	.99	.98	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.86	.85	.84	.83	.83	.82
20	259	.99	.99	.98	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.86	.85	.84	.83	.83	.82
40	287	1.00	.99	.99	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.86	.85	.84	.84	.83	.82
60	308	1.00	.99	.99	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.86	.85	.84	.84	.83	.82
80	324	1.00	1.00	.99	.99	.98	.97	.96	.94	.93	.92	.91	.90	.89	.88	.87	.86	.86	.85	.84	.84	.83	.82
100	338	–	1.00	1.00	.99	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.85	.84	.83	.82
120	350	–	1.00	1.00	.99	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.85	.84	.83	.82
140	361	–	–	1.00	1.00	.99	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.85	.84	.83	.82
160	371	–	–	–	1.00	.99	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.83	.82
180	380	–	–	–	1.00	.99	.98	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.83	.83
200	388	–	–	–	1.00	.99	.99	.97	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.83	.83
220	395	–	–	–	1.00	1.00	.99	.98	.96	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.84	.83
240	403	–	–	–	–	1.00	.99	.98	.97	.95	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.84	.83
260	409	–	–	–	–	1.00	.99	.98	.97	.96	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.85	.84	.83
280	416	–	–	–	–	1.00	1.00	.99	.97	.96	.95	.93	.92	.91	.90	.89	.88	.87	.86	.85	.85	.84	.83
300	422	–	–	–	–	–	1.00	.99	.98	.96	.95	.93	.92	.91	.90	.89	.88	.87	.86	.86	.85	.84	.83
350	436	–	–	–	–	–	1.00	1.00	.99	.97	.96	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.83
400	448	–	–	–	–	–	–	1.00	.99	.98	.96	.95	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84	.84
450	460	–	–	–	–	–	–	–	1.00	.99	.97	.96	.94	.93	.92	.91	.89	.88	.87	.86	.86	.84	.84
500	470	–	–	–	–	–	–	–	1.00	.99	.98	.96	.94	.93	.92	.91	.90	.89	.88	.87	.86	.85	.84
550	480	–	–	–	–	–	–	–	–	1.00	.99	.97	.95	.94	.92	.91	.90	.89	.88	.87	.86	.85	.84
600	489	–	–	–	–	–	–	–	–	1.00	.99	.98	.96	.94	.93	.92	.90	.89	.88	.87	.86	.85	.84
650	497	–	–	–	–	–	–	–	–	–	1.00	.99	.97	.95	.94	.92	.91	.90	.89	.87	.86	.86	.85
700	506	–	–	–	–	–	–	–	–	–	1.00	.99	.97	.96	.94	.93	.91	.90	.89	.88	.87	.86	.85
750	513	–	–	–	–	–	–	–	–	–	1.00	1.00	.98	.96	.95	.93	.92	.90	.89	.88	.87	.86	.85
800	520	–	–	–	–	–	–	–	–	–	–	1.00	.99	.97	.95	.94	.92	.91	.90	.88	.87	.86	.85
850	527	–	–	–	–	–	–	–	–	–	–	1.00	.99	.98	.96	.94	.93	.92	.90	.89	.88	.87	.86
900	533	–	–	–	–	–	–	–	–	–	–	1.00	1.00	.99	.97	.95	.93	.92	.90	.89	.88	.87	.86
950	540	–	–	–	–	–	–	–	–	–	–	–	1.00	.99	.97	.95	.94	.92	.91	.89	.88	.87	.86
1000	546	–	–	–	–	–	–	–	–	–	–	–	1.00	.99	.98	.96	.94	.93	.91	.90	.89	.87	.86

Table 3: Properties of Saturated Steam

Gauge Pressure PSIG	Temper-ature °F	Heat in Btu/lb. Sensible	Latent	Total	Specific Volume Cu. ft. per lb.	Gauge Pressure PSIG	Temper-ature °F	Heat in Btu/lb. Sensible	Latent	Total	Specific Volume Cu. ft. per lb.
IN VAC. **25**	**134**	**102**	**1017**	**1119**	**142.0**	185	382	355	843	1198	2.29
20	162	129	1001	1130	73.9	190	384	358	841	1199	2.24
15	179	147	990	1137	51.3	195	386	360	839	1199	2.19
10	192	160	982	1142	39.4	**200**	**388**	**362**	**837**	**1199**	**2.14**
5	203	171	976	1147	31.8	205	390	364	836	1200	2.09
0	**212**	**180**	**970**	**1150**	**26.8**	210	392	366	834	1200	2.05
1	215	183	968	1151	25.2	215	394	368	832	1200	2.00
2	219	187	966	1153	23.5	220	396	370	830	1200	1.96
3	222	190	964	1154	22.3	**225**	**397**	**372**	**828**	**1200**	**1.92**
4	224	192	962	1154	21.4	230	399	374	827	1201	1.89
5	**227**	**195**	**960**	**1155**	**20.1**	235	401	376	825	1201	1.85
6	230	198	959	1157	19.4	240	403	378	823	1201	1.81
7	232	200	957	1157	18.7	245	404	380	822	1202	1.78
8	233	201	956	1157	18.4	**250**	**406**	**382**	**820**	**1202**	**1.75**
9	237	205	954	1159	17.1	255	408	383	819	1202	1.72
10	**239**	**207**	**953**	**1160**	**16.5**	260	409	385	817	1202	1.69
12	244	212	949	1161	15.3	265	411	387	815	1202	1.66
14	248	216	947	1163	14.3	270	413	389	814	1203	1.63
16	252	220	944	1164	13.4	**275**	**414**	**391**	**812**	**1203**	**1.60**
18	256	224	941	1165	12.6	280	416	392	811	1203	1.57
20	**259**	**227**	**939**	**1166**	**11.9**	285	417	394	809	1203	1.55
22	262	230	937	1167	11.3	290	418	395	808	1203	1.53
24	265	233	934	1167	10.8	295	420	397	806	1203	1.49
26	268	236	933	1169	10.3	**300**	**421**	**398**	**805**	**1203**	**1.47**
28	271	239	930	1169	9.85	305	423	400	803	1203	1.45
30	**274**	**243**	**929**	**1172**	**9.46**	310	425	402	802	1204	1.43
32	277	246	927	1173	9.10	315	426	404	800	1204	1.41
34	279	248	925	1173	8.75	320	427	405	799	1204	1.38
36	282	251	923	1174	8.42	**325**	**429**	**407**	**797**	**1204**	**1.36**
38	284	253	922	1175	8.08	330	430	408	796	1204	1.34
40	**286**	**256**	**920**	**1176**	**7.82**	335	432	410	794	1204	1.33
42	289	258	918	1176	7.57	340	433	411	793	1204	1.31
44	291	260	917	1177	7.31	345	434	413	791	1204	1.29
46	293	262	915	1177	7.14	**350**	**435**	**414**	**790**	**1204**	**1.28**
48	295	264	914	1178	6.94	355	437	416	789	1205	1.26
50	**298**	**267**	**912**	**1179**	**6.68**	360	438	417	788	1205	1.24
55	300	271	909	1180	6.27	365	440	419	786	1205	1.22
60	307	277	906	1183	5.84	370	441	420	785	1205	1.20
65	312	282	901	1183	5.49	**375**	**442**	**421**	**784**	**1205**	**1.19**
70	316	286	898	1184	5.18	380	443	422	783	1205	1.18
75	**320**	**290**	**895**	**1185**	**4.91**	385	445	424	781	1205	1.16
80	324	294	891	1185	4.67	390	446	425	780	1205	1.14
85	328	298	889	1187	4.44	395	447	427	778	1205	1.13
90	331	302	886	1188	4.24	**400**	**448**	**428**	**777**	**1205**	**1.12**
95	335	305	883	1188	4.05	450	460	439	766	1205	1.00
100	**338**	**309**	**880**	**1189**	**3.89**	500	470	453	751	1204	.89
105	341	312	878	1190	3.74	550	479	464	740	1204	.82
110	344	316	875	1191	3.59	600	489	473	730	1203	.75
115	347	319	873	1192	3.46	**650**	**497**	**483**	**719**	**1202**	**.69**
120	350	322	871	1193	3.34	700	505	491	710	1201	.64
125	**353**	**325**	**868**	**1193**	**3.23**	750	513	504	696	1200	.60
130	356	328	866	1194	3.12	800	520	512	686	1198	.56
135	358	330	864	1194	3.02	900	534	529	666	1195	.49
140	361	333	861	1194	2.92	**1000**	**546**	**544**	**647**	**1191**	**.44**
145	363	336	859	1195	2.84	1250	574	580	600	1180	.34
150	**366**	**339**	**857**	**1196**	**2.74**	1500	597	610	557	1167	.23
155	368	341	855	1196	2.68	1750	618	642	509	1151	.22
160	371	344	853	1197	2.60	2000	636	672	462	1134	.19
165	373	346	851	1197	2.54	**2250**	**654**	**701**	**413**	**1114**	**.16**
170	375	348	849	1197	2.47	2500	669	733	358	1091	.13
175	**377**	**351**	**847**	**1198**	**2.41**	2750	683	764	295	1059	.11
180	380	353	845	1198	2.34	3000	696	804	213	1017	.08

Draining Steam Mains

Steam main drainage is one of the most common applications for steam traps. It is important that water is removed from steam mains as quickly as possible, for reasons of safety and to permit greater plant efficiency. A build-up of water can lead to waterhammer, capable of fracturing pipes and fittings. When carried into the steam spaces of heat exchangers, it simply adds to the thickness of the condensate film and reduces heat transfer. Inadequate drainage leads to leaking joints, and is a potential cause of wire-drawing of control valve seats.

Waterhammer

Waterhammer occurs when a slug of water, pushed by steam pressure along a pipe instead of draining away at the low points, is suddenly stopped by impact on a valve or fitting such as a pipe bend or tee. The velocities which such slugs of water can achieve are not often appreciated. They can be much higher than the normal steam velocity in the pipe, especially when the waterhammer is occurring at startup.

When these velocities are destroyed, the kinetic energy in the water is converted into pressure energy and a pressure shock is applied to the obstruction. In mild cases, there is noise and perhaps movement of the pipe. More severe cases lead to fracture of the pipe or fittings with almost explosive effect, and consequent escape of live steam at the fracture.

Waterhammer is avoided completely if steps are taken to ensure that water is drained away before it accumulates in sufficient quantity to be picked up by the steam.

Careful consideration of steam main drainage can avoid damage to the steam main and possible injury or even loss of life. It offers a better alternative than an acceptance of waterhammer and an attempt to contain it by choice of materials, or pressure rating of equipment.

Efficient Steam Main Drainage

Proper drainage of lines, and some care in start up methods, not only prevent damage by waterhammer, but help improve steam quality, so that equipment output can be maximized and maintenance of control valves reduced.

The use of oversized steam traps giving very generous “safety factors” does not necessarily ensure safe and effective steam main drainage. A number of points must be kept in mind, for a satisfactory installation.

1) The heat up method employed.
2) Provision of suitable collecting legs or reservoirs for the condensate.
3) Provision of a minimum pressure differential across the steam trap.
4) Choice of steam trap type and size.
5) Proper trap installation.

Heat Up Method

The choice of steam trap depends on the heat up method adopted to bring the steam main up to full pressure and temperature. The two most usual methods are:

(a) supervised start up and
(b) automatic start up.

A) Supervised Start Up

In this case, at each drain point in the steam system, a manual drain valve is fitted, bypassing the steam trap and discharging to atmosphere.

These drain valves are opened fully before any steam is admitted to the system. When the “heat up” condensate has been discharged and as the pressure in the main begins to rise, the valves are closed. The condensate formed under operating conditions is then discharged through the traps. Clearly, the traps need only be sized to handle the losses from the lines under operating conditions, given in Table 5 (page 10).

This heat up procedure is most often used in large installations where start up of the system is an infrequent, perhaps even an annual, occurrence. Large heating systems and chemical processing plants are typical examples.

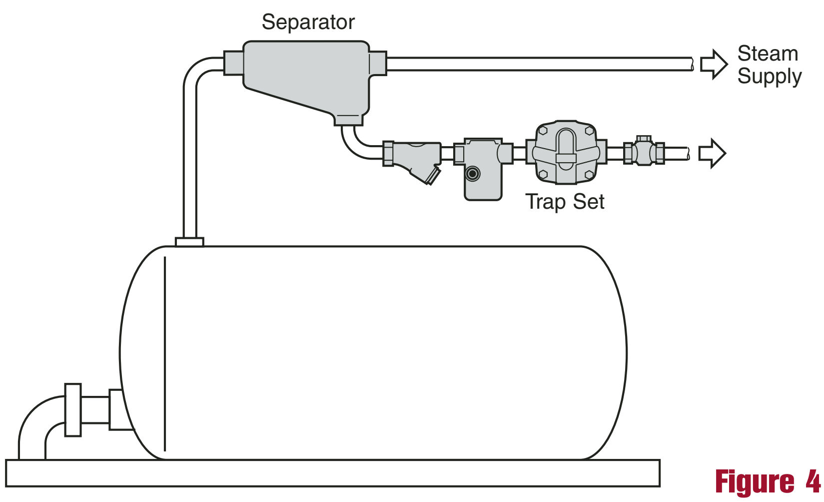


Figure 4
Trap Boiler header or takeoff separator and size for maximum carryover. On heavy demand this could be 10% of generating capacity

B) Automatic Start Up

One traditional method of achieving automatic start up is simply to allow the steam boiler to be fired and brought up to pressure with the steam take off valve (crown valve) wide open. Thus the steam main and branch lines come up to pressure and temperature without supervision, and the steam traps are relied on to automatically discharge the condensate as it is formed.

This method is generally confined to small installations that are regularly and frequently shut down and started up again. For example, the boilers in many laundry and drycleaning plants are often shut down at night and restarted the next morning.

In anything but the smallest plants, the flow of steam from the boiler into the cold pipes at start up, while the boiler pressure is still only a few psi, will lead to excessive carryover of boiler water with the steam. Such carryover can be enough to overload separators in the steam takeoff, where these are fitted. Great care, and even good fortune, are needed if waterhammer is to be avoided.

For these reasons, modern practice calls for an automatic valve to be fitted in the steam supply line, arranged so that the valve stays closed until a reasonable pressure is attained in the boiler. The valve can then be made to open over a timed period so that steam is admitted only slowly into the distribution pipework. The pressure with the boiler may be climbing at a fast rate, of course, but the slow opening valve protects the pipework.

Where these valves are used, the time available to warm up the pipework will be known, as it is set on the valve control. In other cases it is necessary to know the details of the boiler start up procedure so that the time can be estimated. Boilers started from cold are often fired for a short time and then shut off while temperatures equalize. The boilers are protected from undue stress by these short bursts of firing, which extend the warmup time and reduce the rate at which condensation in the mains is to be discharged at the traps.

Determining Condensate Loads

As previously discussed there are two methods for bringing a steam main "on line". The supervised start up bypasses the traps thus avoiding the large warm up loads. The traps are then sized based on the running conditions found in Table 5 (page 10). A safety factor of 2:1 and a differential pressure of inlet minus condensate return pressure.

Systems employing automatic start up procedures requires estimation of the amount of condensate produced in bringing up the main to working temperature and pressure within the time available. The amount of condensate being formed and the pressure available to discharge it are both varying continually and at any given moment are indeterminate due to many unknown variables. Table 4 (page 10) indicates the warm up loads per 100 feet of steam main during a one hour start up. If the start up time is different, the new load can be calculated as follows:

$$\frac{\text{lbs. of Condensate (Table 4)} \times 60}{\text{Warm up time in minutes}} = \text{Actual warm-up load.}$$

Apply a safety factor of 2:1 and size the trap at a differential pressure of working steam pressure minus condensate return line presure. Since most drip traps see running loads much more often than start up loads, care must be taken when sizing them for start up conditions. If the start up load forces the selection of a trap exceeding the capability of the "running load trap," then the warm up time needs to be increased and/or the length of pipe decreased.

Warm Up Load Example

Consider a length of 8" main which is to carry steam at 125 psig. Drip points are to be 150 ft. apart and outside ambient conditions can be as low as 0°F. Warm-up time is to be 30 minutes.

From Table 4, Warm Up Load is 107 lb./100 ft.

For a 150 ft run, load is 107 x 1.5 = 160.5 lb/150 ft.

Correction Factor for 0°F (see Table 4) 1.25 x 160.5 = 200.6 lb/150 ft.

A 30 minute warm up time increases the load by

$$\frac{200.6 \times 60}{30} = 401 \text{ lb/h total load}$$

Applying a safety factor of 2:1, the trap sizing load is 802 lb/h. If the back pressure in the condensate return is 0 psig, the trap would be sized for a 125 psi differential pressure. This would result in an oversized trap during running conditions, calculated at 94 lb/h using Tabe 5 (page 10). Either increase the warm up time to one hour or decrease the distance between drip traps.

Draining Steam Mains

Table 4: Warm-Up Load in Pounds of Steam per 100 Ft of Steam Main

Ambient Temperature 70°F. Based on Sch. 40 pipe to 250 psi, Sch. 80 above 250 except Sch. 120 5" and larger above 800 psi

Steam Pressure psi	2"	2½"	3"	4"	5"	6"	8"	10"	12"	14"	16"	18"	20"	24"	0°F Correction Factor†
0	6•2	9•7	12•8	18•2	24•6	31•9	48	68	90	107	140	176	207	308	1•50
5	6•9	11•0	14•4	20•4	27•7	35•9	48	77	101	120	157	198	233	324	1•44
10	7•5	11•8	15•5	22•0	29•9	38•8	58	83	109	130	169	213	251	350	1•41
20	8•4	13•4	17•5	24•9	33•8	44	66	93	124	146	191	241	284	396	1•37
40	9•9	15•8	20•6	90•3	39•7	52	78	110	145	172	225	284	334	465	1•32
60	11•0	17•5	22•9	32•6	44	57	86	122	162	192	250	316	372	518	1•29
80	12•0	19•0	24•9	35•3	48	62	93	132	175	208	271	342	403	561	1•27
100	12•8	20•3	26•6	37•8	51	67	100	142	188	222	290	366	431	600	1•26
125	13•7	21•7	28•4	40	55	71	107	152	200	238	310	391	461	642	1•25
150	14•5	23•0	30•0	43	58	75	113	160	212	251	328	414	487	679	1•24
175	15•3	24•2	31•7	45	61	79	119	169	224	265	347	437	514	716	1•23
200	16•0	25•3	33•1	47	64	83	125	177	234	277	362	456	537	748	1•22
250	17•2	27•3	35•8	51	69	89	134	191	252	299	390	492	579	807	1•21
300	25•0	38•3	51	75	104	143	217	322	443	531	682	854	1045	1182	1•20
400	27•8	43	57	83	116	159	241	358	493	590	759	971	1163	1650	1•18
500	30•2	46	62	91	126	173	262	389	535	642	825	1033	1263	1793	1•17
600	32•7	50	67	98	136	187	284	421	579	694	893	1118	1367	1939	1•16
800	38	58	77	113	203	274	455	670	943	1132	1445	1835	2227	3227	1•156
1000	45	64	86	126	227	305	508	748	1052	1263	1612	2047	2485	3601	1•147
1200	52	72	96	140	253	340	566	833	1172	1407	1796	2280	2767	4010	1•140
1400	62	79	106	155	280	376	626	922	1297	1558	1988	2524	3064	4440	1•135
1600	71	87	117	171	309	415	692	1018	1432	1720	2194	2786	3382	4901	1•130
1750	78	94	126	184	333	448	746	1098	1544	1855	2367	3006	3648	5285	1•128
1800	80	97	129	189	341	459	764	1125	1584	1902	2427	3082	3741	5420	1•127

†For outdoor temperature of 0°F, multiply load value in table for each main size by correction factor shown.

Table 5: Running Load in Pounds per Hour per 100 Ft of Insulated Steam Main

Ambient Temperature 70°F. Insulation 80% efficient. Load due to radiation and convection for saturated steam.

Steam Pressure psi	2"	2½"	3"	4"	5"	6"	8"	10"	12"	14"	16"	18"	20"	24"	0°F Correction Factor†
10	6	7	9	11	13	16	20	24	29	32	36	39	44	53	1•58
30	8	9	11	14	17	20	26	32	38	42	48	51	57	68	1•50
60	10	12	14	18	24	27	33	41	49	54	62	67	74	89	1•45
100	12	15	18	22	28	33	41	51	61	67	77	83	93	111	1•41
125	13	16	20	24	30	36	45	56	66	73	84	90	101	121	1•39
175	16	19	23	26	33	38	53	66	78	86	98	107	119	142	1•38
250	18	22	27	34	42	50	62	77	92	101	116	126	140	168	1•36
300	20	25	30	37	46	54	68	85	101	111	126	138	154	184	1•35
400	23	28	34	43	53	63	80	99	118	130	148	162	180	216	1•33
500	27	33	39	49	61	73	91	114	135	148	170	185	206	246	1•32
600	30	37	44	55	68	82	103	128	152	167	191	208	232	277	1•31
800	36	44	53	69	85	101	131	164	194	214	244	274	305	365	1•30
1000	43	52	63	82	101	120	156	195	231	254	290	326	363	435	1•27
1200	51	62	75	97	119	142	185	230	274	301	343	386	430	515	1•26
1400	60	73	89	114	141	168	219	273	324	356	407	457	509	610	1•25
1600	69	85	103	132	163	195	253	315	375	412	470	528	588	704	1•22
1750	76	93	113	145	179	213	278	346	411	452	516	580	645	773	1•22
1800	79	96	117	150	185	221	288	358	425	467	534	600	667	800	1•21

†For outdoor temperature of 0°F, multiply load value in table for each main size by correction factor shown.

Draining Steam Mains

Note from the example that in most cases, other than large distribution mains, 1/2" Thermo-Dynamic® traps have ample capacity. For shorter lengths between drip points, and for small diameter pipes, the 1/2" low capacity TD trap more than meets even start up loads, but on larger mains it may be worth fitting parallel 1/2" traps as in Fig. II-6 (page 86). Low pressure mains are best drained using float and thermostatic traps, and these traps can also be used at higher pressures.

The design of drip stations are fairly simple. The most common rules to follow for sizing the drip pockets are:

1. The diameter of the drip pockets shall be the same size as the distribution line up to 6 inches in diameter. The diameter shall be half the size of the distribution line over 6 inches but never less than 6 inches.
2. The length of the drip pocket shall be 1-1/2 times the diameter of the distribution line but not less than 18 inches.

Drip Leg Spacing

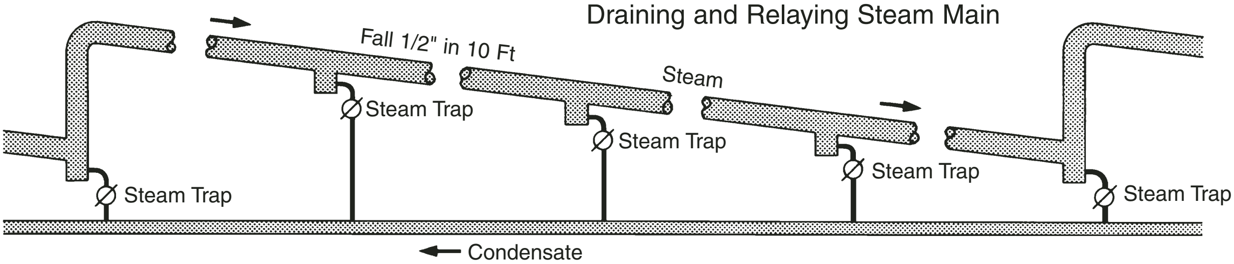
The spacing between the drainage points is often greater than is desirable. On a long horizontal run (or rather one with a fall in the direction of the flow of about 1/2" in 10 feet or 1/250) drain points should be provided at intervals of 100 to 200 feet. Longer lengths should be split up by additional drain points. Any natural collecting points in the systems, such as at the foot of any riser, should also be drained.

A very long run laid with a fall in this way may become so low that at intervals it must be elevated with a riser. The foot of each of these "relay points" also requires a collecting pocket and steam trap.

Sometimes the ground contours are such that the steam main can only be run uphill. This will mean the drain points should be at closer intervals, say 50 ft. apart, and the size of the main increased. The lower steam velocity then allows the condensate to drain in the opposite direction to the steam flow.

Air venting of steam mains is of paramount importance and is far too often overlooked. Steam entering the pipes tends to push the air already there in front of it as would a piston. Automatic air vents, fitted on top of tees at the terminal points of the main and the larger branches, will allow discharge of this air. Absence of air vents means that the air will pass through the steam traps (where it may well slow down the discharge of condensate) or through the steam using equipment itself.

Figure 5
Draining and Relaying Steam Main



Case in Action: Steam Main and Steam Tracing System Drainage

The majority of steam traps in refineries are installed on steam main and steam tracing systems. Thorough drainage of steam mains/branch lines is essential for effective heat transfer around the refinery and for waterhammer prevention. This holds true for condensate drainage from steam tracing lines/jackets, though some degree of backup (or sub-cooling) is permissible in some applications.

The predominant steam trap installed is a non-repairable type that incorporates a permanent pipeline connector. Scattered throughout the system are a number of iron and steel body repairable types.

Most notable failure of steam traps are precipitate formation on bucket weep-holes and discharge orifices that eventually plugs the trap shut. A common culprit is valve sealing compound injected into leaking valves which forms small pellets that settle in low points, such as drip legs/steam traps and on strainer screens making blow down difficult. This problem also occurs during occasional "system upset" when hydrocarbon contaminants are mistakenly introduced to the steam system.

A noise detector and/or a temperature-indicating device is required to detect trap failure. Especially costly is the fact that operators are not allowed to remove traps for repair when threading from the line is required. Maintenance personnel must be involved.

Solution

Universal connector steam traps were installed for trial in one of the dirtiest drip stations at the refinery. The traps held up under adverse operating conditions requiring only periodic cleaning. Since the time of installation, all failed inverted bucket traps in this service were replaced with universal connector traps. Strainers were installed upstream of each.

Benefits

- The addition of Thermo-Dynamic® traps allowed for easier field trap testing.
- The addition of universal connectors significantly reduced steam trap installation and repair time.
- 33% reduction in steam trap inventory due to standard trap for all sizes.
- Reduced energy loss is significantly reduced using Thermo-Dynamic® steam traps versus original inverted bucket traps.

The temperature of process liquids being transferred through pipelines often must be maintained to meet the requirements of a process, to prevent thickening and solidification, or simply to protect against freezeup. This is achieved by the use of jacketed pipes, or by attaching to the product line one or more separate tracer lines carrying a heating medium such as steam or hot water.

The steam usage may be relatively small but the tracing system is often a major part of the steam installation, and the source of many problems.

Many large users and plant contractors have their own inhouse rules for tracer lines, but the following guidelines may be useful in other cases. We have dealt only with external tracing, this being the area likely to cause difficulties where no existing experience is available. External tracing is simple and therefore cheap to install, and fulfills the needs of most processes.

External Tracer Lines

One or more heat carrying lines, of sizes usually from 3/8" up to 1" nominal bore are attached to the main product pipe as in Fig. 6. Transfer of heat to the product line may be three ways—by conduction through direct contact, by convection currents in the air pocket formed inside the insulating jacket, and by radiation. The tracer lines may be of carbon steel or copper, or sometimes stainless steel.

Where the product line is of a particular material to suit the fluid it is carrying, the material for the tracer line must be chosen to avoid electrolytic corrosion at any contact points.

For short runs of tracer, such as around short vertical pipes, or valves and fittings, small bore copper pipes, perhaps 1/4" bore may be wound around the product lines as at Fig. 7. The layout should be arranged to give a continuous fall along the tracers as Fig. 9a rather than Fig. 9b, and the use of wrap around tracers should be avoided on long horizontal lines.

A run of even 100 ft. of 6 inch product line will have a total of about 500 to 600 ft. of wrap around tracer. The pressure drop along the tracer would be very high and the temperature at the end remote from the supply would be very low. Indeed, this end of the tracer would probably contain only condensate and the temperature of this water would fall as it gives up heat. Where steam is present in the tracer, lifting the condensate from the multiplicity of low points increases the problems associated with this arrangement.

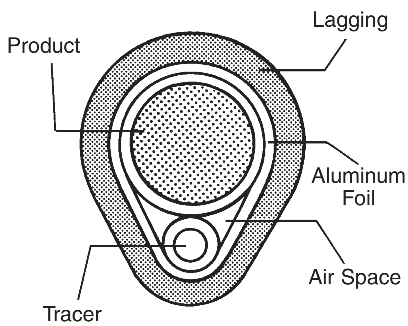


Figure 6
Tracer Attached To Product Line

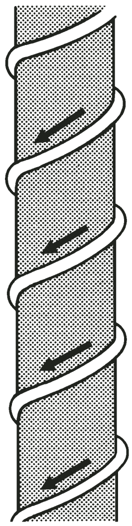


Figure 7
Small Bore Tracing Wraped Around Vertical Product Line

Figure 9 Continuous Fall On Wrap Around Tracer

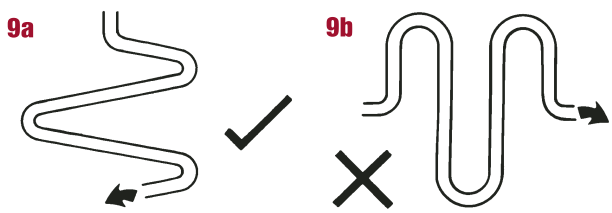


Figure 10 Attaching Tracer To Line

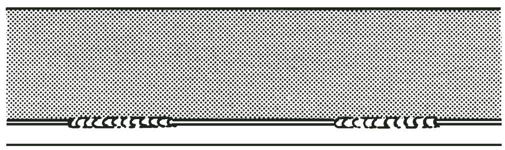


Figure 10a Short Run Welds

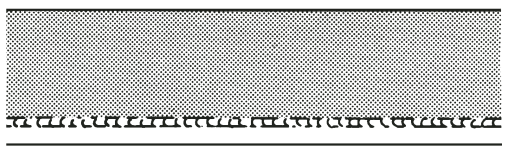


Figure 10b Continuous Weld

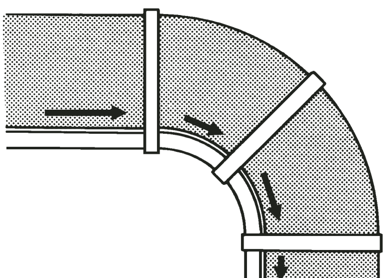


Figure 8
Clipping Tracer Around Bends

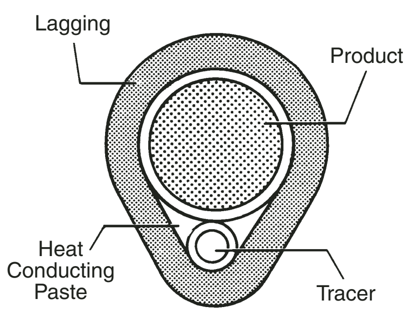


Figure 10c Heat Conducting Paste

Clip On Tracers

The simplest form of tracer is one that is clipped or wired on to the main product line. Maximum heat flow is achieved when the tracer is in tight contact with the product line. The securing clips should be no further apart than 12" to 18" on 3/8" tracers, 18" to 24 on 1/2", and 24" to 36" on 3/4" and larger.

The tracer pipes can be literally wired on, but to maintain close contact it is better to use either galvanized or stainless steel bands, about 1/2" wide and 18 to 20 gauge thickness. One very practical method is to use a packing case banding machine. Where tracers are carried around bends particular care should be taken to ensure that good contact is maintained by using three or more bands as in Fig. 8.

Where it is not possible to use bands as at valve bodies, soft annealed stainless steel wire 18 gauge thick is a useful alternative.

Once again, any special needs to avoid external corrosion or electrolytic action may lead to these suggestions being varied.

Welded Tracers

Where the temperature difference between the tracer and the product is low, the tracer may be welded to the product line. This can be done either by short run welds as Fig. 10a or by a continuous weld as Fig. 10b for maximum heat transfer.

In these cases the tracer is sometimes laid along the top of the pipe rather than at the bottom, which greatly simplifies the welding procedure. Advocates of this method claim that this location does not adversely affect the heat transfer rates.

Heat Conducting Paste

For maximum heat transfer, it can be an advantage to use a heat conducting paste to fill the normal hot air gap as in Fig. 10c. The paste can be used to improve heat transfer with any of the clipping methods described, but it is essential that the surfaces are wirebrushed clean before applying the paste.

Spacer Tracing

The product being carried in the line can be sensitive to temperature in some cases and it is then important to avoid any local hot spots on the pipe such as could occur with direct contact between the tracer and the line.

This is done by introducing a strip of insulating material between the tracer and the product pipe such as fiberglass, mineral wool, or packing blocks of an inert material.

Insulation

The insulation must cover both the product line and the tracer but it is important that the air space remains clear. This can be achieved in more than one way.

1. The product line and tracer can first be wrapped with aluminum foil, or by galvanized steel sheet, held on by wiring and the insulation is then applied outside this sheet. Alternatively, small mesh galvanized wire netting can be used in the same way as metal sheet Fig. 11a.
2. Sectional insulation, preformed to one or two sizes larger than the product main, can be used. This has the disadvantage that it can easily be crushed Fig. 11b.
3. Preformed sectional insulation designed to cover both product line and tracer can be used, as Fig. 11c.

Preformed sectional insulation is usually preferred to plastic material, because being rigid it retains better thickness and efficiency. In all cases, the insulation should be properly finished with waterproof covering. Most insulation is porous and becomes useless as heat conserving material if it is allowed to absorb water. Adequate steps may also be needed to protect the insulation from mechanical damage.

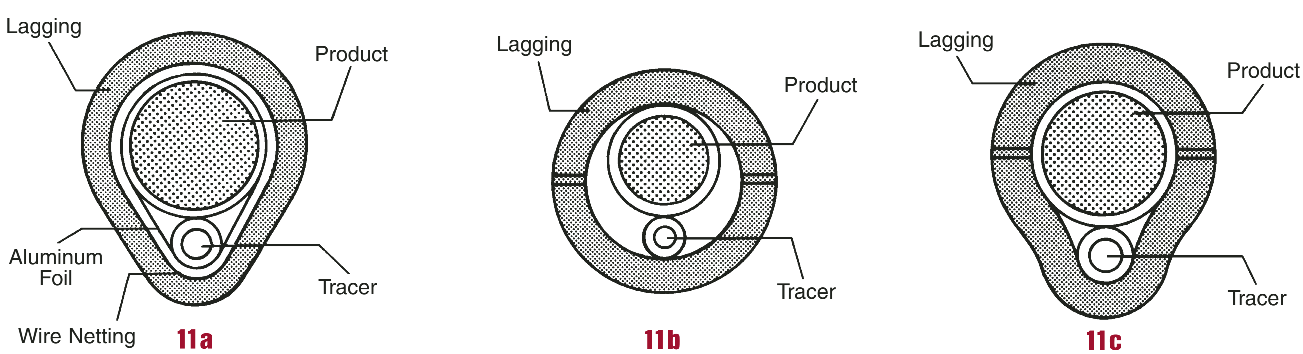


Figure 11
Insulating Tracer and Product Lines

Sizing of External Tracers

The tracing or jacketing of any line normally aims at maintaining the contents of the line at a satisfactory working temperature under all conditions of low ambient temperature with adequate reserve to meet extreme conditions.

Remember that on some exposed sites, with an ambient still air temperature of say 0°F, the effect of a 15 mph wind will be to lower the temperature to an equivalent of -36°F.

Even 32°F in still air can be lowered to an effective 4°F with a 20 mph wind—circumstances which must be taken into full consideration when studying the tracer line requirements.

Details of prevailing conditions can usually be obtained from the local meteorological office or civil air authority.

Most of the sizing of external tracers is done by rule of thumb, but the problem which arises here is what rule and whose thumb?

Rules of thumb are generally based on the experiences of a certain company on a particular process and do not necessarily apply elsewhere. There are also widely differing opinions on the layout: some say that multiple tracers should all be below the center line of the product line while others say, with equal conviction, that it is perfectly satisfactory to space the tracers equally around the line.

Then there are those who will endeavor to size their tracers from 3/8", 1/2", 3/4" or 1" and even larger pipe: while another school of thought says that as tracers have only minute contact with the product line it will give much more even distribution of heat if all tracers are from 1/2" pipe in multiples to meet the requirements. This does have the added advantage of needing to hold a stock of only one size of pipe and fittings rather than a variety of sizes.

For those who like to follow this idea, Table 6 will be useful for most average requirements.

Type A would suffice for most fuel oil requirements and would also meet the requirement of those lines carrying acid, phenol, water and some other chemicals, but in some cases spacers between the product line and steam line would be employed.

The steam pressure is important and must be chosen according to the product temperature required.

For noncritical tracing Types A & B (Table 6) a steam pressure of 50 psi would generally be suitable. For Type C, a higher pressure and a trap with a hot discharge may be required.

Table 6: Number of 1/2" (15mm) Tracers Used with Different Sizes of Product Lines

	Type A Noncritical General frost protection or where solidification may occur at temps below 75°F	**Type B Noncritical** Where solidification may occur at temps between 75-150°F	**Type C Critical** When solidification may occur at temps between 150-300°F
Product Line Size	**Number of 1/2" Tracers**	**Number of 1/2" Tracers**	**Number of 1/2" Tracers**
1"	1	1	1
1½"	1	1	2
2"	1	1	2
3"	1	1	3
4"	1	2	3
6"	2	2	3
8"	2	2	3
10"-12"	2	3	6
14"-16"	2	3	8
18"-20"	2	3	10

Jacketed Lines

Ideally jacketed lines should be constructed in no more than 20 ft. lengths and the condensate removed from each section. Steam should enter at the highest end so that there is a natural fall to the condensate outlet as Fig. 12a.

When it is considered impractical to trap each length, a number of lengths up a total of 80-100 ft. approx. may be joined together in moderate climates, but in extremely cold parts of the world 40 ft. should be the maximum. See Fig. 12b.

Always avoid connecting solely through the bottom loop. This can only handle the condensate and impedes the free flow of steam as Fig. 12c. As a general guide, see Table 7.

Although in most cases 1/2" condensate outlet will be adequate, it is usual to make this the same size as the steam connection as it simplifies installation.

External Tracers

In horizontal runs, the steam will generally flow parallel to the product line, but as far as possible, steam should enter from the high end to allow free flow of the condensate to the low end, i.e. it should always be self-draining.

It is generally considered preferable to fit one tracer on the bottom of the line as Fig. 13a, two tracers at 30° as Fig. 13b, three tracers at 45° as Fig. 13c.

Where multiple 1/2" tracers are used, they should be arranged in loop fashion on either side of the product line, as Fig. 14. In vertical lines, the tracers would be spaced uniformly, as Fig. 15a & b.

The maximum permissible length of tracer will depend to some extent on the size and initial steam pressure, but as a general guide 3/8" tracers should not exceed 60 ft. in length and the limit for all other sizes should be about 150 ft.

Bends and low points in the tracer, as Fig. 16a should always be avoided. For example, if it is necessary to carry a tracer line round a pipe support or flange,

this should be done in the horizontal plane, Fig. 16b.

Where it is essential to maintain the flow of heat to the product, the tracer should be taken up to the back of the flange Fig. 17, and the coupling should always be on the center line of the flanged joint.

The same applies to an in-line run where the tracer has to be jointed. This can be done in two ways, Fig. 18 or Fig. 19.

Each of these is preferable to Fig. 20 which could produce a cold spot. Where two tracers are used it can be better to double back at a union or flange as Fig. 21, rather than jump over it.

Expansion

Expansion in tracer lines is often overlooked. Naturally the steam heated tracer will tend to expand more than the product line. Where the tracer has to pass around flanges, the bends are quite adequate to take care of the expansion, Fig. 22.

But where this does not occur and there is a long run of uninterrupted tracer, it is essential to provide for expansion which can be done by forming a complete loop, Fig. 23.

Table 7: Steam Connection Size for Jacketed Lines

Product Line	Jacket Diameter	Steam Connection
2-1/2" 65mm	4" 100mm	1/2" 15mm
3" 80mm	6" 150mm	3/4" 20mm
4" 100mm	6" 150mm	3/4" 20mm
6" 150mm	8" 200mm	3/4" 20mm
8" 200mm	10" 250mm	1" 25mm
10" 250mm	12" 300mm	1" 25mm

General Installation

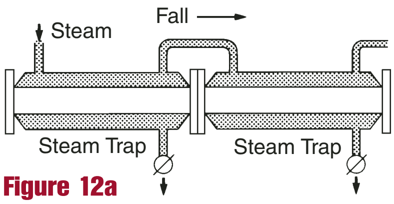


Figure 12a
Jacketed Lines, Drained Separately

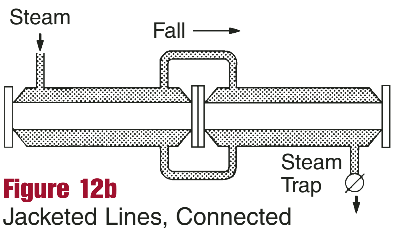


Figure 12b
Jacketed Lines, Connected

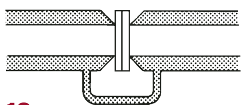
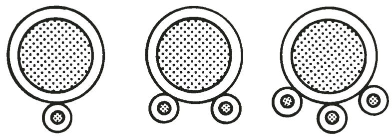


Figure 12c
Incorrect Arrangement of Jacketed Lines

Figure 13
Single and Multiple Tracing



13a **13b** **13c**

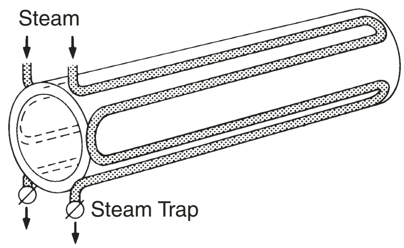
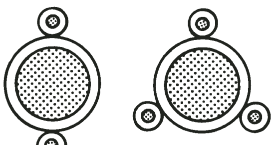


Figure 14
Multiple Tracing

Figure 15 Vertical Tracing



15a **15b**

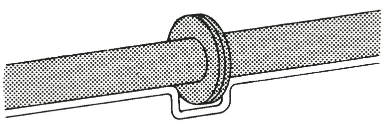


Figure 16a Incorrect Arrangement

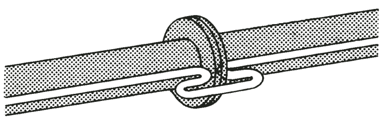


Figure 16b Correct Arrangement

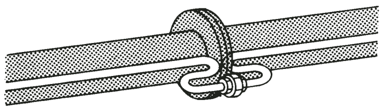


Figure 17

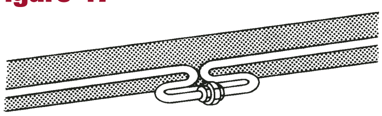


Figure 18

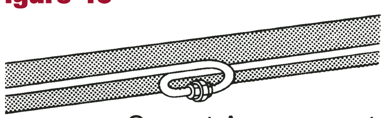


Figure 19 Correct Arrangement for Tracer-line Joints

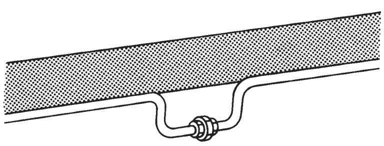


Figure 20 Incorrect Arrangement

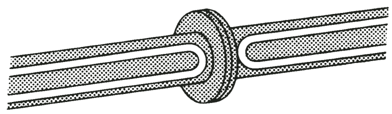


Figure 21
Dual Tracer Double Back

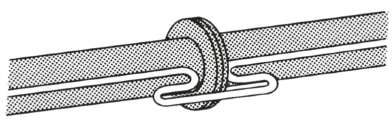


Figure 22

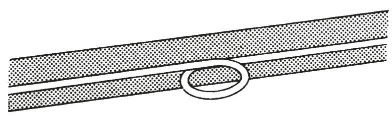


Figure 23
Expansion Arrangements on Long Tracers

Tracer Steam Distribution

It is important that the steam supply should always be taken from a source which is continuously available, even during a normal shut down period.

Tracer lines and jacketed pipe may have to work at any steam pressure (usually in the range between 10 and 250 psi, but always choose the lowest pressure to give the required product temperature. Excessively high pressures cause much waste and should only be used where a high product temperature is essential).

To suit product temperature requirements, it may be necessary to use steam at different pressures. It should be distributed at the highest pressure and reduced down to meet the lower pressure requirements. A Reducing Valve can be used for this purpose, Fig. 24. Note: it may be necessary to steam trace the valve body to prevent damage due to freezing..

A number of tracers can be supplied from one local distribution header. This header should be adequately sized to meet the maximum load and drained at its low point by a steam trap as Fig. 25. All branches should be taken off the top of this header, one branch to each tracer line. These branches should be fitted with isolating valves.

Don't undersize these branch connections (1/2" supply to even a 3/8" tracer will avoid undue pressure drop) and serve only tracers local to the header, otherwise high pressure drop may result.

The size of the header will, of course, depend upon the steam pressure and the total load on the tracers but as a general guide, see Table 8:

Table 8

Recommended header size for supplying steam tracer lines	
Header Size	**Number of 1/2" Tracers**
3/4"	2
1"	3-5
1½"	6-15
2"	16-30
Recommended header size for condensate lines	
Header Size	**Number of 1/2" Tracers**
1"	Up to 5
1½"	6-10
2"	11-25

Tracer Trap Sizing

Subcooled discharge traps are usually a good choice for tracer service. Tracing loads are approximately 10 to 50 lb./hr., and each tracer requires its own low capacity trap.

No two tracers can have exactly the same duty, so group trapping two or more tracers to one trap can considerably impair the efficiency of heat transfer, see Fig. 26 and Fig. 27.

Even with multiple tracers on a single product line, each tracer should be separately trapped—Fig. 28.

When branched tracers are taken to serve valves, then each should be separately trapped, Figs. 29, 30, 31 and 32.

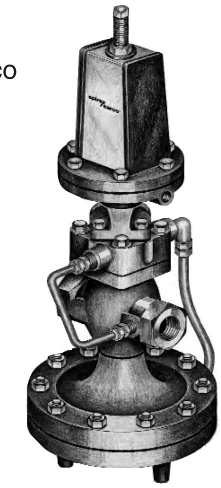


Figure 24
Spirax Sarco Reducing Valve

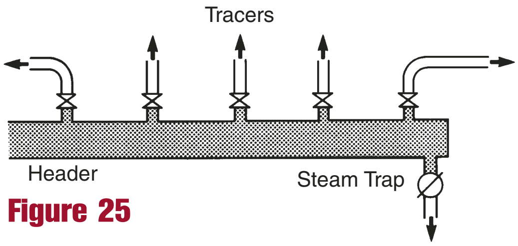


Figure 25

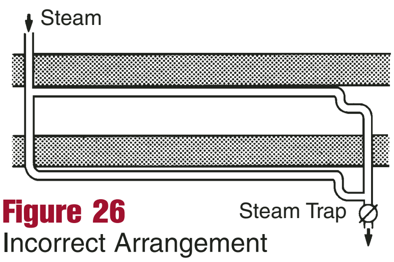


Figure 26
Incorrect Arrangement

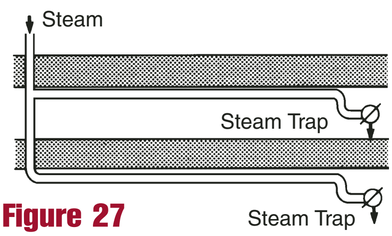


Figure 27
Correct Arrangement

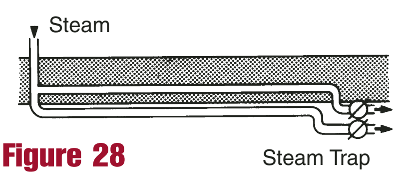


Figure 28

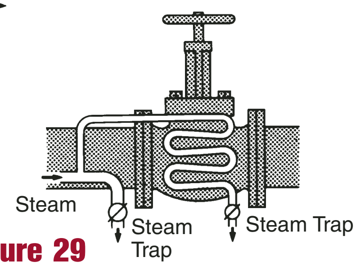


Figure 29

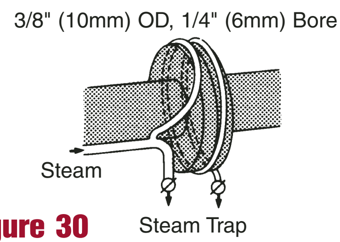


Figure 30

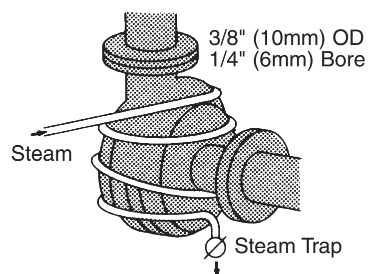


Figure 31
Tracer Lines Around Pump Casing

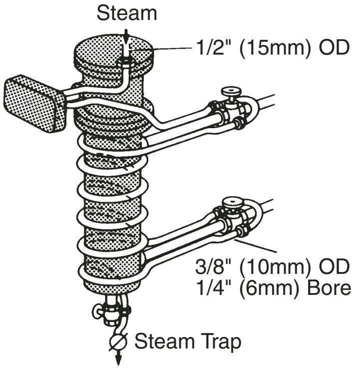


Figure 32
Typical Instrument Tracing

Important– Getting Rid of the Muck

Pipes delivered to the site may contain mill scale, paint, preserving oils, etc. and during storage and erection will collect dirt, sand, weld splatter and other debris, so that on completion, the average tracer line contains a considerable amount of "muck."

Hydraulic testing will convert this "muck" into a mobile sludge which is not adequately washed out by simply draining down after testing.

It is most important that the lines are properly cleaned by blowing through with steam to an open end before diverting to the steam traps.

Unless this is done, the traps will almost certainly fail to operate correctly and more time will be spent cleaning them out when the plant is commissioned.

Steam Traps For Tracer Lines

Almost any type of steam trap could be used to drain tracer lines, but some lend themselves to this application better than others. The traps should be physically small and light in weight, and as they are often fitted in exposed positions, they should be resistant to frost. The temperature at which the condensate is discharged by the trap is perhaps the most important consideration when selecting the type of trap.

Thermo-Dynamic® traps are the simplest and most robust of all traps, they meet all of the above criteria and they discharge condensate at a temperature close to that of steam. Thus they are especially suitable on those tracing applications where the holding back of condensate in the tracer line until it has subcooled would be unacceptable. Tracers or jackets on lines carrying sulphur or asphalt typify these applications where the tracer must be at steam temperature along its whole length.

It must be remembered that every time a Thermo-Dynamic® trap opens, it discharges condensate at the maximum rate corresponding to the differential pressure applied. The instantaneous release rates of the steam flashing off the condensate can be appreciable, and care is needed to ensure that condensate return lines are adequately sized if high back pressures are to be avoided. Thus, the use of swept back or "y" connections from trap discharges into common headers of generous size will help avoid problems.

Where the traps are exposed to wind, rain or snow, or low ambient temperatures, the steam bubbles in the top cap of the trap can condense more quickly, leading to more rapid wear. Special insulating caps are available for fitting to the top caps to avoid this, Fig. 33.

In other non-critical applications, it can be convenient and energy efficient to allow the condensate to sub-cool within the tracer before being discharged. This enables use to be made of some of the sensible heat in the condensate, and reduces or even eliminates the release of flash steam. Temperature sensitive traps are then selected, using either balanced pressure or bimetallic elements.

The bimetallic traps usually discharge condensate at some fairly constant differential such as 50°F below condensing temperatures, and tend to give a continuous dribble of condensate when handling tracer loads, helping minimize the size of condensate line needed. They are available either in maintainable versions, with a replaceable element set which includes the valve and seat as well as the bimetallic stack, or as sealed non-maintainable units as required.

Balanced pressure traps normally operate just below steam temperature, for critical tracing applications, see Fig. 34.

The trap is especially suitable where small quantities of condensate are produced, on applications where sub-cooling is desirable, and where the condensate is not to be returned to the recovery system.

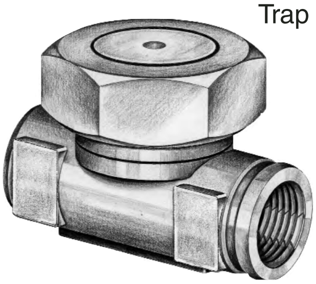


Figure 33
Insulating Cap for Thermo-Dynamic® Trap

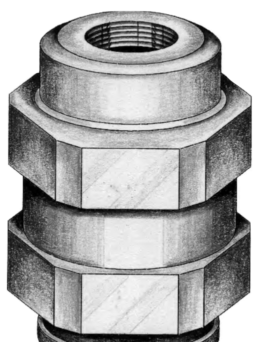


Figure 34
Balanced Pressure Tracer Trap

A similar but maintainable type intended for use on instrument tracer lines, where the physical size of the trap is important as well as its operating characteristics is shown in Fig. 35.

Just as the distribution of steam is from a common header, it often is convenient to connect a number of traps to a common condensate header and this simplifies maintenance. As noted, the discharge should preferably enter the header through swept connections and the headers be adequately sized as suggested in Table 8 (page 16).

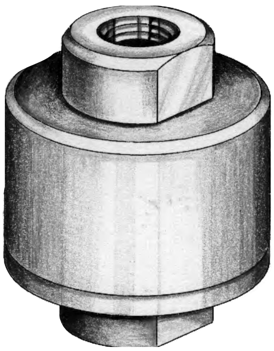


Figure 35
Maintainable Balanced Pressure Tracer Trap.

These may be increased where high pressures and traps discharging condensate at near steam temperature are used, or decreased with low pressures and traps discharging cooler condensate.

Temperature Control of Tracer

Where it is essential to prevent overheating of the product, or where constant viscosity is required for instrumentation, automatic temperature control is frequently used.

On many systems, the simplest way to achieve control is to use a reducing valve on the steam supply to the tracer lines or jacket. This can be adjusted in the light of experience to give the correct steam pressure to produce the required product temperature.

Clearly this is an approximate way to control product temperature and can only be used where the product flow is fairly constant. Where closer control is required, the simple direct acting temperature control often provides an economic solution. This will give close control and since it is not necessary to provide either electric power or compressed air, the first cost and indeed the running costs are low.

Case in Action: Product Steam Tracing with Temperature Control and Overheat Protection

During steam tracing project design, it was found that five thousand feet of 2" product piping was to be traced with 150 psig steam. Product temperature was to be maintained at 100°F, with maximum allowable temperature of 150°F and a minimum allowable temperature of 50°F.

Of particular concern was the fact that the pipeline would always be full of the product, but flow would be intermittent. Overheating could be a real problem. In addition, the tracing system had to be protected from freezing.

Solution

The 5,000 feet of product piping was divided into 30 separate traced sections including: a cast steel temperature regulator, a bronze temperature control valve used as a high limit safety cutout, a sealed balanced pressure thermostatic steam trap, a vacuum breaker, and pressure regulators supplying steam to all 30 tracing sections. Each section operates effectively at the desired temperature, regardless of flow rate or ambient temperature.

Benefits

- The chance of product damage from overheating is minimized and steam consumption is reduced through steam pressure reduction (150 psig to 50 psig) with the pressure regulator.
- The product temperature is maintained at a consistent set temperature, maximizing process control under all flow conditions with the temperature regulator.
- Product damage from overheating is prevented through use of the high limit safety cutout. The system will shut down completely, should the temperature regulator overshoot its set point.
- The tracing system is protected from freezing with the sealed balanced pressure thermostatic steam trap discharging to drain. Thorough drainage is also facilitated by the vacuum breaker.

Pressure Reducing Stations

It is a mistake to install even the best of pressure reducing valves in a pipeline without giving some thought to how best it can be helped to give optimal performance.

The valve selected should be of such a size that it can handle the necessary load, but oversizing should be avoided. The weight of steam to be handled in a given time must be calculated or estimated, and a valve capable of passing this weight from the given upstream pressure to the required downstream pressure is chosen. The valve size is usually smaller than the steam pipes either upstream or downstream, because of the high velocities which accompany the pressure drop within the valve.

Types of Pressure Reducing Valves are also important and can be divided into three groups of operation as follows:

Direct Operated Valves

The direct acting valve shown in Fig. II-17 (page 91) is the simplest design of reducing valve.

This type of valve has two drawbacks in that it allows greater fluctuation of the downstream pressure under unstable load demands, and these valves have relatively low capacity for their size. It is nevertheless perfectly adequate for a whole range of simple applications where accurate control is not essential and where the steam flow is fairly small and constant.

Pilot Operated Valves

Where accurate control of pressure or large capacity is required, a pilot operated reducing valve should be used. Such a valve is shown in Fig. II-12 (page 89).

The pilot operated design offers a number of advantages over the direct acting valve. Only a very small amount of steam has to flow through the pilot valve to pressurize the main diaphragm chamber and fully open the main valve. Thus, only very small changes in downstream pressure are necessary to produce large changes in flow. The "droop" of pilot operated valves is therefore small. Although any rise in upstream pressure will apply an increased closing force on the main valve, this is offset by the force of the upstream pressure acting on the main diaphragm. The result is a valve which gives close control of downstream pressure regardless of variations on the upstream side.

Pneumatically Operated Valves

Pneumatically operated control valves, Fig. II-20 (page 93), with actuators and positioners being piloted by controllers, will provide pressure reduction with even more accurate control.

Controllers sense downstream pressure fluctuations, interpolate the signals and regulate an air supply signal to a pneumatic positioner which in turn supplies air to a disphragm opening a valve. Springs are utilized as an opposing force causing the valves to close upon a loss or reduction of air pressure applied on the diaphragm. Industry sophistication and control needs are demanding closer and more accurate control of steam pressures, making pneumatic control valves much more popular today.

Piping And Noise Consideration

The piping around a steam pressure reducing valve must be properly sized and fitted for best operation. Noise level of a reducing station is lowest when the valve is installed as follows:

1. Avoid abrupt changes in direction of flow. Use long radius bends and "Y" piping instead of "T" connections.
2. Limit approach and exit steam velocity to 4000 to 6000 FPM.
3. Change piping gradually before and after the valve with tapered expanders, or change pipe only 1 or 2 sizes at a time.
4. Provide long, straight, full-size runs of heavy wall pipe on both sides of the valve, and between two-stage reductions to stabilize the flow.
5. Use low pressure turndown ratios (non-critical.)
6. Install vibration absorbing pipe hangers and acoustical insulation.

Most noise is generated by a reducing valve that operates at critical pressure drop, especially with high flow requirements. Fitting a noise diffuser directly to the valve outlet will reduce the noise level by approx. 15 dBA.

It must also be remembered that a valve designed to operate on steam should not be expected to work at its best when supplied with a mixture of steam, water and dirt.

A separator, drained with a steam trap, will remove almost all the water from the steam entering the pressure reducing set. The baffle type separator illustrated in Fig. 36 has been found to be very effective over a broad range of flows.

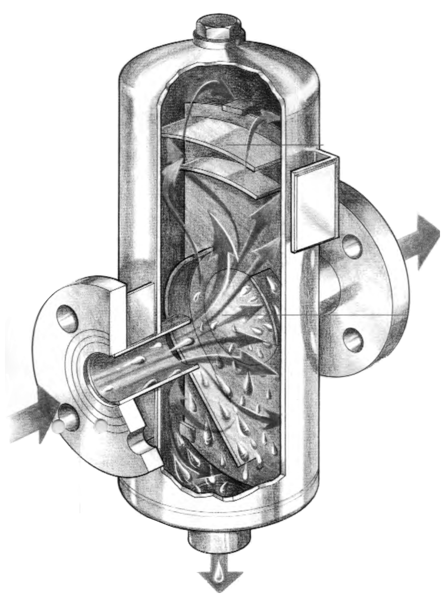


Figure 36
Moisture Separator for Steam or Air

PRV Station Components

A stop valve is usually needed so that the steam supply can be shut off when necessary, and this should be followed by a line size strainer. A fine mesh stainless steel screen in the strainer will catch the finer particles of dirt which pass freely through standard strainers. The strainer should be installed in the pipe on its side, rather than in the conventional way with the screen hanging below the pipe. This is to avoid the screen space acting as a collecting pocket for condensate, since when installed horizontally the strainer can be self-draining

Remember that water which collects in the conventionally piped strainer at times when the reducing valve has closed, will be carried into the valve when it begins to open. This water, when forced between the valve disc and seat of the just-opening valve, can lead rapidly to wire-drawing, and the need for expensive replacements.

Pressure gauges at each side of the reducing valve allow its performance to be monitored. At the reduced pressure side of the valve, a relief or safety valve may be required. If all the equipment connected on the low pressure side is capable of safely withstanding the upstream pressure in the event of reducing valve failure, the relief valve may not be needed. It may be called for if it is sought to protect material in process from overly high temperatures, and it is essential if any downstream equipment is designed for a pressure lower than the supply pressure.

Steam Safety Valve Sizing

When selecting a safety valve, the pressure at which it is to open must be decided. Opening pressure must be below the limitations of the downstream equipment yet far enough above the normal reduced pressure that minor fluctuations do not cause opening or dribbling. Type “UV” Safety Valves for unfired pressure vessels are tested to ASME Pressure Vessel Code, Section VIII and achieve rated capacity at an accumulated pressure 10% above the set-to-open pressure. Safety valves for use on boilers carry a “V” stamp and achieve rated capacity at only 3% overpressure as required by Section I of the Code.

The capacity of the safety valve must then equal or exceed the capacity of the pressure reducing valve, if it should fail open when discharging steam from the upstream pressure to the accumulated pressure at the safety valve. Any bypass line leakage must also be accounted for.

Figure 37
Typical Installation of Single Reducing Valve with Noise Diffuser

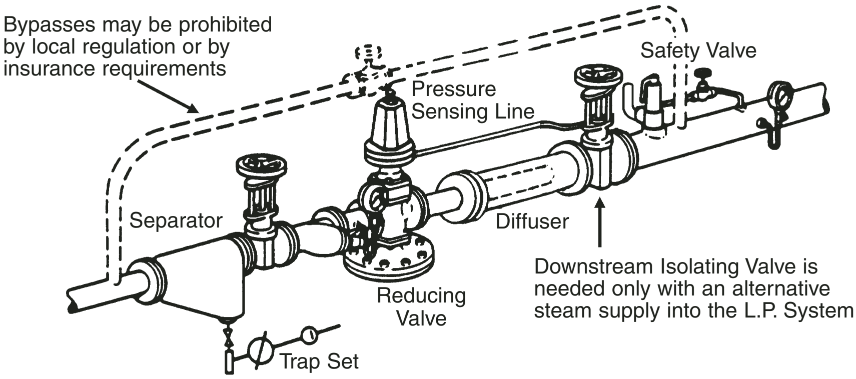
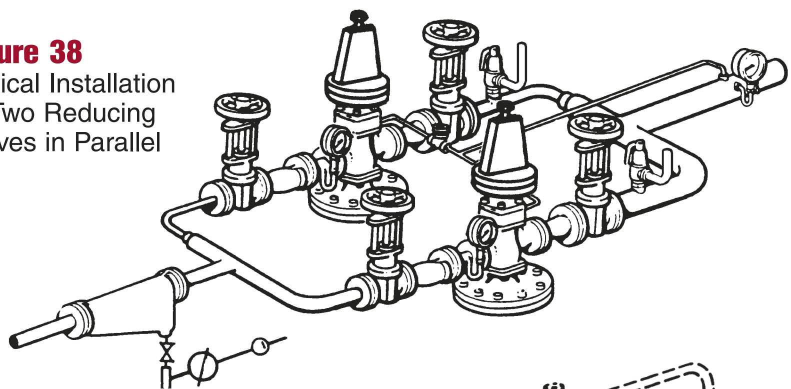


Figure 38
Typical Installation of Two Reducing Valves in Parallel



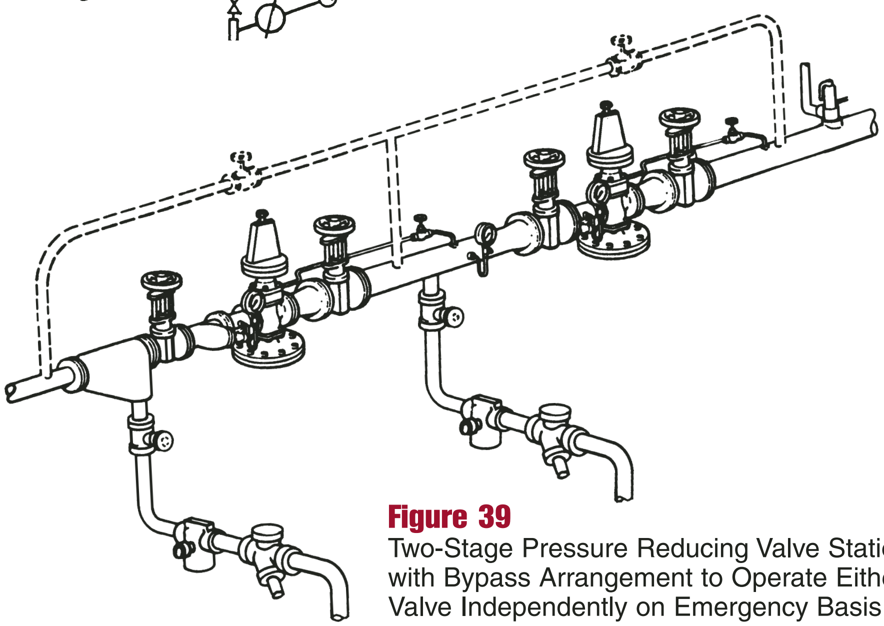


Figure 39
Two-Stage Pressure Reducing Valve Station with Bypass Arrangement to Operate Either Valve Independently on Emergency Basis

Case in Action: Elimination of Steam Energy Waste

As part of a broad scope strategy to reduce operating costs throughout the refinery, a plan was established to eliminate all possible steam waste. The focus of the plan was piping leaks, steam trap failures and steam pressure optimization.

Programs having been previously established to detect/repair steam trap failures and fix piping leaks, particular emphasis was placed on steam pressure optimization. Results from a system audit showed that a considerable amount of non-critical, low temperature tracing was being done with 190 psi (medium pressure) steam, an expensive overkill. It appeared that the medium pressure header had been tapped for numerous small tracing projects over the years.

Solution

Refinery engineers looked for ways to reduce pressure to the tracer lines. Being part of a cost-cutting exercise, it had to be done without spending large sums of capital money on expensive control valves. The self-contained cast steel pressure regulators and bronze reducing valves were chosen for the job. In 1-1/2 years, approximately 40 pressure regulators and hundreds of bronze reducing valves have been installed at a cost of $250K. Annualized steam energy savings are $1.2M/year. More specifically, in the Blending and Shipping Division, $62,640 was saved during the winter of 1995, compared to the same period in 1994.

Benefits

- Low installed cost. The Spirax Sarco regulators and bronze reducing valves are completely self-contained, requiring no auxiliary controllers, positioners, converters, etc.
- Energy savings worth an estimated $1.2M/year.
- The utilities supervisor who worked closely with Spirax Sarco and drove the project through to successful completion received company wide recognition and a promotion in grade.

Parallel Operation

In steam systems where load demands fluctuate through a wide range, multiple pressure control valves with combined capacities meeting the maximum load perform better than a single, large valve. Maintenance needs, downtime and overall lifetime cost can all be minimized with this arrangement, Fig. 38 (page 20).

Any reducing valve must be capable of both meeting its maximum load and also modulating down towards zero loads when required. The amount of load turndown which a given valve can satisfactorily cover is limited, and while there are no rules which apply without exception, if the low load condition represents 10% or less of the maximum load, two valves should always be preferred. Consider a valve which moves away from the seat by 0.1 inches when a downstream pressure 1 psi below the set pressure is detected, and which then passes 1,000 pounds per hour of steam. A rise of 0.1 psi in the detected pressure then moves the valve 0.01 inches toward the seat and reduces the flow by approximately 100 pph, or 10%.

The same valve might later be on a light load of 100 pph total when it will be only 0.01 inches away from the seat. A similar rise in the downstream pressure of 0.1 psi would then close the valve completely and the change in flow through the valve which was 10% at the high load, is now 100% at low load. The figures chosen are arbitrary, but the principle remains true that instability or “hunting” is much more likely on a valve asked to cope with a high turndown in load.

A single valve, when used in this way, tends to open and close, or at least move further open and further closed, on light loads. This action leads to wear on both the seating and guiding surfaces and reduces the life of the diaphragms which operate the valve. The situation is worsened with those valves which use pistons sliding within cylinders to position the valve head. Friction and sticking between the sliding surfaces mean that the valve head can only be moved in a series of discreet steps. Especially at light loads, such movements are likely to result in changes in flow rate which are grossly in excess of the load changes which initiate them. Load turndown ratios with piston-operated valves are almost inevitably smaller than where diaphragm-operated valves are chosen.

Pressure Settings for Parallel Valves

Automatic selection of the valve or valves needed to meet given load conditions is readily achieved by setting the valves to control at pressures separated by one or two psi. At full load, or loads not too much below full load, both valves are in use. As the load is reduced, the controlled pressure begins to increase and the valve set at the lower pressure modulates toward the closed position. When the load can be supplied completely by the valve set at the higher pressure, the other valve closes and with any further load reduction, the valve still in use modulates through its own proportional band.

Parallel and Series Operation of Reducing Valves

This can be clarified by an example. Suppose that a maximum load of 5,000 lb/h at 30 psi can be supplied through one valve capable of passing 4,000 lb/h and a parallel valve capable of 1,300 lb/h. One valve is set at 29 psi and the other at 31 psi. If the smaller valve is the one set at 31 psi, this valve is used to meet loads from zero up to 1,300 lb/h with a controlled pressure at approximately 31 psi. At greater loads, the controlled pressure drops to 29 psi and the larger valve opens, until eventually it is passing 3,700 lb/h to add to the 1,300 lb/h coming through the smaller valve for a total of 5,000 lb/h.

There may be applications where the load does not normally fall below the minimum capacity of the larger valve. It would then be quite normal to set the 4,000 lb/h valve at 31 psi and to supplement the flow through the 1,300 lb/h valve at 29 psi in those few occasions when the extra capacity was required.

Sometimes the split between the loads is effectively unknown. It is usual then to simply select valves with capacities of 1/3 and 2/3 of the maximum with the smaller valve at the slightly higher pressure and the larger one at the slightly lower pressure.

Two-Stage or Series Operation

Where the total reduction in pressure is through a ratio of more than 10 to 1, consideration should be given to using two valves in series, Fig. 39 (page 20). Much will depend on the valves being used, on the total pressure reduction needed and the variations in the load. Pilot Operated controls have been used successfully with a pressure turndown ratio as great as 20 to 1, and could perhaps be used on a fairly steady load from 100 psig to 5 psi. The same valve would probably be unstable on a variable load, reducing from 40 to 2 psi.

There is no hard and fast rule, but two valves in series will usually provide more accurate control. The second, or Low Pressure valve, should give the "fine control" with a modest turndown, with due consideration being given to valve sizes and capacities. A practical approach when selecting the turndown of each valve, that results in smallest most economical valves, is to avoid having a non-critical drop in the final valve, and stay close to the recommended 10 to 1 turndown.

Series Installations

For correct operation of the valves, some volume between them is needed if stability is to be achieved. A length of 50 pipe diameters of the appropriately sized pipe for the intermediate pressure, or the equivalent volume of larger diameter pipe is often recommended.

It is important that the downstream pressure sensing pipes are connected to a straight section of pipe 10 diameters downstream from the nearest elbow, tee, valve or other obstruction. This sensing line should be pitched to drain away from the pressure pilot. If it is not possible to arrange for this and to still connect into the top of the downstream pipe, the sensing line can often be connected to the side of the pipe instead.

Equally, the pipe between the two reducing valves should always be drained through a stream trap, just as any riser downstream of the pressure reducing station should be drained. The same applies where a pressure reducing valve supplies a control valve, and it is essential that the connecting pipe is drained upstream of the control valve.

Bypasses

The use of bypass lines and valves should usually be avoided. Where they are fitted, the capacity through the bypass should be added to that through the wide open reducing valve when sizing relief valves. Bypass valves are often found to be leaking steam because of wiredrawing of the seating faces when valves have not been closed tightly.

If a genuine need exists for a bypass because it is essential to maintain the supply of steam, even when a reducing valve has developed some fault or is undergoing maintenance, consideration should be given to fitting a reducing valve in the bypass line. Sometimes the use of a parallel reducing station of itself avoids the need for bypasses.

Back Pressure Controls

A Back Pressure regulator or surplussing valve is a derivative of a pressure reducing valve, incorporating a reverse acting pilot valve. The pressure sensing pipe is connected to the inlet piping so that the pilot valve responds to upstream pressure. Any increase in upstream pressure then opens the reverse acting pilot valve, causing the main valve to open, while a fall below the set pressure causes the main valve to close down, Fig. II-18 (page 92).

These controls are useful in flash steam recovery applications when the supply of flash steam may at times exceed the demand for it. The BP control can then surplus to atmosphere any excess steam tending to increase the pressure within the flash steam recovery system, and maintains the recovery pressure at the required level.

The control is also useful in eliminating non-essential loads in any system that suffers undercapacity at peak load times, leaving essential loads on line.

Back Pressure Controls are not Safety Valves and must never be used to replace them.

How to Size Temperature and Pressure Control Valves

Having determined the heating or cooling load required by the equipment, a valve must be selected to handle it. As the valve itself is only part of the complete control, we must be acquainted with certain terminology used in the controls field:

Flow Coefficient. The means of comparing the flow capacities of control valves by reference to a "coefficient of capacity." The term Cv is used to express this relationship between pressure drop and flow rate. Cv is the rate of flow of water in GPM at 60°F, at a pressure drop of 1 psi across the fully open valve.

Differential Pressure. The difference in pressure between the inlet and outlet ports when the valve is closed. For three-port valves, it is the difference between the open and closed ports.

Maximum Differential Pressure. The pressure difference between inlet and outlet ports of a valve, above which the actuator will not be able to close the valve fully, or above which damage may be caused to the valve, whichever is the smaller.

Pressure Drop. The difference between the inlet and outlet pressures when the valve is passing the stated quantity. A self-acting control may or may not be fully open. For three-port valves, it is the difference in pressure between the two open ports.

Working Pressure. The pressure exerted on the interior of a valve under normal working conditions. In water systems, it is the algebraic sum of the static pressure and the pressure created by pumps.

Set Point. Pressure or temperature at which controller is set.

Accuracy of Regulation or "Droop". Pressure reducing valve drop in set point pressure necessary to obtain the published capacity. Usually stated for pilot-operated PRV's in psi, and as a % of set pressure for direct-acting types.

Hunting or Cycling. Persistent periodic change in the controlled pressure or temperature.

Control Point. Actual value of the controlled variable (e.g. air temperature) which the sensor is trying to maintain.

Deviation. The difference between the set point and the measured value of the controlled variable. (Example: When set point is 70°F and air temperature is 68°F, the deviation is 2°F.)

Offset. Sustained deviation caused by a proportional control taking corrective action to satisfy a load condition. (Example: If the set point is 70°F and measured room temperature is 68°F over a period, the offset is 2°F and indicates the action of a proportional control correcting for an increase in heat loss.)

Proportional Band or Throttling Band. Range of values which cause a proportional temperature control to move its valve from fully open to fully closed or to throttle the valve at some reduced motion to fully closed.

Time Constant. Time required for a thermal system actuator to travel 63.2% of the total movement resulting from any temperature change at the sensor. Time increase when using separable well must be included.

Dead Zone. The range of values of the controlled variable over which a control will take up no corrective action.

Rangeability. The ratio between the maximum and minimum controllable flow between which the characteristics of the valve will be maintained.

Turn-Down Ratio. The ratio between the maximum normal flow and minimum controllable flow.

Valve Authority. Ratio of a fully open control valve pressure drop to system total pressure drop.

Case in Action: Log Bath-Furniture Manufacturing

At a furniture manufacturing facility, the water used for bathing logs to prepare them for production was "rolling" in the front of its containment tanks. The production manager had thought that the temperature had to be at least 212 °F. Further examination showed the water's temperature to be 180°F. The water was "rolling" because the steam, entering the side of the tank, could not be absorbed by the water before it rose to the surface in the front of the tank.

Cedar logs are cooked for 48 hours, in open top tanks before going through a veneer machine. The logs absorb the hot water, making it easier to slice the wood into strips. The six log baths did not have any temperature controls. Twenty-five psig steam flowed through a 2" coupling into the side of the tank to heat the water. With the tank size being 12' x 12' x 6', the 105 cedar logs approximately 10' long occupy most of the space in the tank. River water or "condenser water" off of the turbine at 90°F is fed into the tank.

Solution

Two temperature control valves to be open during start-up with one closing as it approaches the desired cooking temperature. The second smaller valve continues to provide steam to the system until the set-point is reached. As additional steam is required, the smaller valve supplies it. A sparge pipe was also sized and installed.

Benefits:

- Payback of this system was less than 2 weeks on materials and labor.
- Substantial cost savings due to improved energy use.
- Increased profitability by increasing productivity in the steam system.

How to Size Temperature and Pressure Control Valves

Calculating Condensate Loads

When the normal condensate load is not known, the load can be approximately determined by calculations using the following formula.

General Usage Formulae

Heating water with steam (Exchangers)*

$$\text{lb/h Condensate} = \frac{\text{GPM x (1.1) x Temperature Rise °F}}{2}$$

Heating fuel oil with steam

$$\text{lb/h Condensate} = \frac{\text{GPM x (1.1) x Temperature Rise °F}}{4}$$

Heating air with steam coils

$$\text{lb/h Condensate} = \frac{\text{CFM x Temperature Rise °F}}{800}$$

Steam Radiation

$$\text{lb/h Condensate} = \frac{\text{Sq. Ft. EDR}}{4}$$

*Delete the (1.1) factor when steam is injected directly into water

Specialized Applications

Sterilizers, Autoclaves, Retorts Heating Solid Material

$$\text{lb/h Condensate} = \frac{W \times Cp \times \Delta T}{L \times t}$$

W = Weight of material—lbs.
Cp = Specific heat of the material
(Δ)T = Temperature rise of the material °F
L = Latent heat of steam Btu/lb
t = Time in hours

Heating Liquids in Steam Jacketed Kettles and Steam Heated Tanks

$$\text{lb/h Condensate} = \frac{G \times s.g. \times Cp \times (\Delta)T \times 8.3}{L \times t}$$

G = Gallons of liquid to be heated
s.g. = Specific gravity of the liquid
Cp = Specific heat of the liquid
(Δ)T = Temperature rise of the liquid °F
L = Latent heat of the steam Btu/lb
t = Time in hours

Heating Air with Steam; Pipe Coils and Radiation

$$\text{lb/h Condensate} = \frac{A \times U \times (\Delta)T}{L}$$

A = Area of the heating surface in square feet
U = Heat transfer coefficient (2 for free convection)
(Δ)T = Steam temperature minus the air temperature °F
L = Latent heat of the steam Btu/lb

Valve Sizing For Steam

Satisfactory control of steam flow to give required pressures in steam lines or steam spaces, or required temperatures in heated fluids, depends greatly on selecting the most appropriate size of valve for the application.

An oversized valve tends to hunt, with the controlled value (pressure or temperature), oscillating on either side of the desired control point. It will always seek to operate with the valve disc nearer to the seat than a smaller valve which has to be further open to pass the required flow. Operation with the disc near to the seat increases the likelihood that any droplets of water in the steam supply will give rise to wiredrawing. An undersized valve will simply unable to meet peak load requirements, startup times will be extended and the steam-using equipment will be unable to provide the required output.

A valve size should not be determined by the size of the piping into which it is to be fitted. A pressure drop through a steam valve seat of even a few psi means that the steam moves through the seat at high velocity. Valve discs and seats are usually hardened materials to withstand such conditions. The velocities acceptable in the piping are much lower if erosion of the pipes themselves is to be avoided. Equally, the pressure drop of a few psi through the valve would imply a much greater pressure drop along a length of pipe if the same velocity were maintained, and usually insufficient pressure would be left for the steam-using equipment to be able to meet the load.

Steam valves should be selected on the basis of the required steam flow capacity (lb/h) needed to pass, the inlet pressure of the steam supply at the valve, and the pressure drop which can be allowed across the valve. In most cases, proper sizing will lead to the use of valves which are smaller than the pipework on either side.

Steam Jacketed Dryers

$$\text{lb/h Condensate} = \frac{1000\,(Wi - Wf) + (Wi \times \Delta T)}{L}$$

Wi = Initial weight of the material—lb/h
Wf = Final weight of the material—lb/h
(Δ)T = Temperature rise of the material °F
L = Latent heat of steam Btu/lb

Note: The condensate load to heat the equipment must be added to the condensate load for heating the material. Use same formula.

Temperature Control Valve Sizing

After estimating the amount of steam flow capacity (lbs/hr) which the valve must pass, decide on the pressure drop which can be allowed. Where the minimum pressure in a heater, which enables it to meet the load, is known, this value then becomes the downstream pressure for the control valve. Where it is not known, it is reasonable to take a pressure drop across the valve of some 25% of the absolute inlet pressure. Lower pressure drops down to 10% can give acceptable results where thermo-hydraulic control systems are used. Greater pressure drops can be used when it is known that the resulting downstream pressure is still sufficiently high. However, steam control valves cannot be selected with output pressures less than 58% of the absolute inlet pressure. This pressure drop of 42% of the absolute pressure is called Critical Pressure Drop. The steam then reaches Critical or Sonic velocity. Increasing the pressure drop to give a final pressure below the Critical Pressure gives no further increase in flow.

Pressure Reducing Valve Sizing

Pressure reducing valves are selected in the same way, but here the reduced or downstream pressure will be specified. Capacity tables will list the Steam Flow Capacity (lb/h) through the valves with given upstream pressures, and varying downstream pressures. Again, the maximum steam flow is reached at the Critical Pressure Drop and this value cannot be exceeded.

It must be noted here that for self-acting regulators, the published steam capacity is always given for a stated "Accuracy of Regulation" that differs among manufacturers and is not always the maximum the PRV will pass. Thus when sizing a safety valve, the C_V must be used.

Cv Values

These provide a means of comparing the flow capacities of valves of different sizes, type or manufacturer. The C_V factor is determined experimentally and gives the GPM of water that a valve will pass with a pressure drop of 1 psi. The C_V required for a given application is estimated from the formulae, and a valve is selected from the manufacturers catalog to have an equal or greater C_V factor.

1. For Liquids

$$C_v = GPM \sqrt{\frac{\text{Sp. Gr.}}{\text{Pressure Drop, psi}}}$$

Where Sp. Gr. Water = 1
GPM = Gallons per minute

2. For Steam (Saturated)

a. Critical Flow

When ΔP is greater than $F_L^2\ (P_1/2)$

$$C_v = \frac{W}{1.83\ F_L P_1}$$

b. Noncritical Flow

When ΔP is less than $F_L^2\ (P_1/2)$

$$C_v = \frac{W}{2.1\sqrt{\Delta P\ (P_1 + P_2)}}$$

Where: P_1 = Inlet Pressure psia
P_2 = Outlet Pressure psia
W = Capacity lb/hr
F_L = Pressure Recovery Factor
(.9 on globe pattern valves for flow to open)
(.85 on globe pattern valves for flow to close)

3. For Air and Other Gases

a. When P_2 is 0.53 P_1 or less,

$$C_v = \frac{\text{SCFH}\sqrt{\text{Sp. Gr.}}}{30.5\ P_1}$$

Where Sp. Gr. of air is 1.
SCFH is Cu. ft. Free Air per Hour at 14.7 psia, and 60°F.

b. When P_2 is greater than 0.53 P_1,

$$C_v = \frac{\text{SCFH}\sqrt{\text{Sp. Gr.}}}{61\sqrt{(P_1 - P_2)\ P_2}}$$

Where Sp. Gr. of air is 1.
SCFH is Cu. Ft. Free Air per Hour at 14.7 psia, and 60°F.

4. Correction for Superheated Steam

The required Valve C_v is the C_v from the formula multiplied by the correction factor.

Correction Factor = 1 + (.00065 x degrees F. superheat above saturation)

Example: With 25°F of Superheat,
Correction Factor
= 1 + (.00065 x 25)
= 1.01625

5. Correction for Moisture Content

Correction Factor = $\sqrt{\text{Dryness Fraction}}$

Example: With 4% moisture,
Correction Factor = $\sqrt{1 - 0.04}$
= 0.98

6. Gas—Correction for Temperature

$$\text{Correction Factor} = \sqrt{\frac{460 + {}^\circ F}{520}}$$

Example: If gas temperature is 150°F,

$$\text{Correction Factor} = \sqrt{\frac{460 + 150}{520}}$$

= 1.083

Temperature Control Valves for Steam Service

Temperature Control Valves For Steam Service

As with pressure reducing valves, temperature control valves can be divided into three groups. Installation of these valves are the same as pressure reducing styles in that adequate protection from dirt and condensate must be used as well as stop valves for shutdown during maintenance procedures. A noise diffuser and/or a safety valve would normally not be used unless a combination pressure reducing and temperature control, is installed. See PRV station components on page 20 for more information.

Direct Operated Valves

The direct operated type as shown in Fig. 40 are simple in design and operation. In these controls, the thrust pin movement is the direct result of a change in temperature at the sensor. This movement is transferred through the capillary system to the valve, thereby modulating the steam flow. These valves may also be used with hot water. Such a simple relationship between temperature changes and valve stem movement enables sensor and valve combinations to give predictable valve capacities for a range of temperature changes. This allows a valve to be selected to operate with a throttling band within the maximum load proportional band. See appropriate technical sheets for specific valve proportional bands.

Choice of Proportional Band is a combination of accuracy and stability related to each application. However, as control accuracy is of primary importance, and as direct operated controls give constant feedback plus minute movement, we can concentrate on accuracy, leaving the controller to look after stability. Generally, to give light load stability, we would not select a Proportional Band below 2°F. Table 9 gives the span of acceptable Proportional Bands for some common heat exchanger applications.

Pilot Operated Valves

Greater steam capacities are obtained using pilot operated valves, along with greater accuracy due to their 6°F proportional band. Only a small amount of steam has to flow through the pilot to actuate the main diaphragm and fully open the valve. Only very small changes of movement within the sensor are necessary to produce large changes in flow. This results in accurate control even if the upstream steam pressure fluctuates.

Both direct and pilot operated valve types are self-contained and do not require an external power source to operate.

Table 9

Acceptable Proportional Bands for Some Common Applications

Application	Proportional Band °F
Domestic Hot Water Heat Exchanger	7-14
Central Hot Water	4-7
Space Heating (Coils, Convectors, Radiators, etc.)	2-5
Bulk Storage	4-18
Plating Tanks	4-11

Case in Action: Dry Coating Process

An office paper product manufacturer uses steam in its process for a dry coating applied to the paper. Using a pocket ventilation system, air is blown across the paper as it moves through the dryer cans.

The original design included inverted bucket type traps on the outlet of the steam coils, but the coils are in overload boxes where outdoor and indoor air mix. The steam supply is on a modulating control with maximum pressure of 150 psi. The steam traps discharge into a common header that feeds to a liquid mover pump. The pump had a safety relief valve on its non-vented receiver.

Problems observed included the inability to maintain desired air temperatures across the machines, high back pressure on the condensate return system, the doors on the coil boxes had to be opened to increase air flows across the coils, paper machine had to be be slowed down to improve dryness, steam consumption was way up, water make-up was up and vent lines were blowing live steam to the atmosphere.

Solution:

A pump trap combination was installed on five of the nine sections using a pressure regulator for motive steam supply reduction to the pumps. Float & thermostatic traps with leak detection devices were also installed for efficiency. Closed doors were then put on the coil boxes.

The back pressure on the return system dropped to an acceptable and reasonable pressure and the steam consumption also dropped. Temperature control was achieved and maintained and production increased from 1,000 feet per minute on some products to 1,600 feet per minute. They switched all five sections of the paper coater to 1" low profile Pressure Powered Pump with cast iron float & thermostatic steam traps. This manufacturer also switched from inverted buckets on heating units to float and thermostatic steam traps with leak detection devices and replaced several electric pumps and the liquid mover with Pressure Powered Pumps. Replaced all 16 inverted bucket traps on paper coater with float and thermostatic steam traps with leak detection devices.

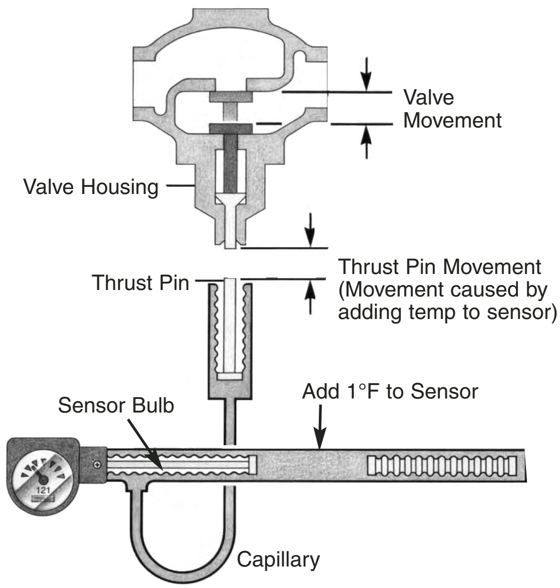
Benefits:

- Production Increased
- Trap failure went from 40% to 14%.
- Over a half-million dollars in steam saved during first year of operation

Pneumatically Operated Valves.

Pneumatically operated temperature control valves as shown in Fig. II-21 (page 93), provide accurate control with the ability to change the setpoint remotely. A controller, through a sensor, adjusts the air signal to the valve actuator or positioner which, in turn, opens or closes the valve as needed. Industry demands for more accurate control of temperature and computer interfacing is making the pneumatically operated valves grow within the marketplace.

Figure 40
Operating Principle of Direct Operated Valves



Installation of Temperature Control Valves

The operation and longevity of these valves depends greatly on the quality of the steam which is fed to them. The components of a temperature control valve station are same as for a pressure reducing valve, see page 19. In addition, attention must be paid to the location of the temperature sensing bulb. It should be completely immersed in the fluid being sensed, with good flow around the bulb, and, if used with a well, some heatsink material in the well to displace the air which prevents heat transfer. The capillary tubing should not be in close proximity to high or low temperatures and should not be crimped in any fashion.

Heating Liquids By Direct Steam Injection

Where noise and dilution of the product are not problems then direct steam injection can be used for heating. Steam injection utilizes all of the latent heat of the steam as well as a large portion of the sensible heat. Two methods, sparge pipes and steam injectors, are used to direct and mix the steam with the product.

Sparge Pipe Sizing

A sparge pipe is simply a perforated pipe used to mix steam with a fluid for heating. Sizing of this pipe is based on determining the required steam flow, selecting a steam pressure within the pipe (normally less than 20 psig for non-pressurized vessels), and calculating the number of holes by dividing the required steam flow by the quantity of steam that will flow through each sparge hole of a specific diameter as determined from Fig. 42. Holes larger than 1/8" diameter are used only on relatively deep tanks where the larger steam bubbles emitted will have time to condense before breaking the liquid surface, or where the required number of 1/8" dia. holes becomes unreasonably great. The sparge holes should be drilled 30° below the horizontal spaced approximately 6" apart and one hole at the bottom to permit drainage of liquid within the pipe, see Fig. 41. The sparge pipe should extend completely across the vessel for complete and even heating.

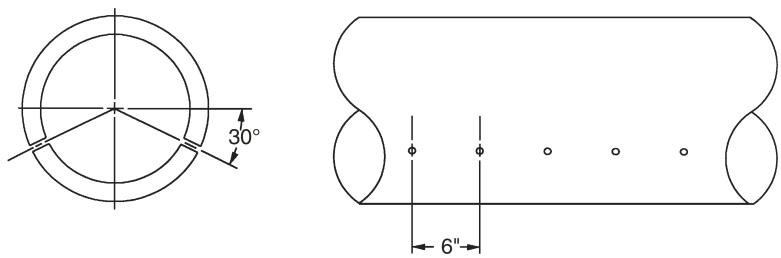


Figure 41
The sparge pipe diameter can be determined using Fig. 1 (page 4), limiting the maximum velocity to 6000 ft/min. A typical installation is shown on Fig II-42 (page 105).

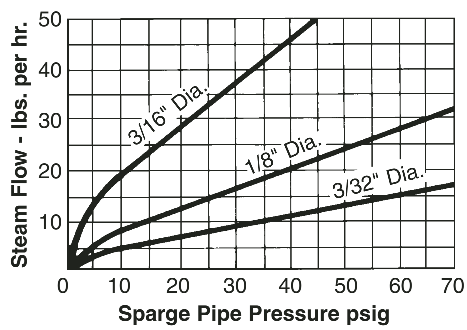


Figure 42
Steam Flow through Sparge Holes

Temperature Control Valves for Steam Service

Steam Injector

Unlike a sparge pipe, a steam injector is a manufactured device that draws in the cold liquid, mixes it with steam within the injector nozzle and distributes the hot liquid throughout the tank. The circulation induced by the injector will help ensure thorough mixing and avoid temperature stratification. See Fig. II-43 (page 105) for a typical injector installation. Other advantages of the injector over a sparge pipe is reduced noise levels and the ability to use high pressure steam up to 200 psig. Refer to applicable technical information sheets for sizing and selection information.

Case in Action: Cheese Production

Fine steam filtration in the preparation of cheese production is important to the quality of the final product. Because the producer of cheese products was heating cheese vat washdown water by direct steam injection, a filtration device was added to enhance product quality by filtering out the particulates.

While pleased with this simple method of heating, there was some concern that any particulates entering the washdown water during steam injection may ultimately contaminate the vats being cleaned, affecting the cheese production.

Solution

Direct steam injection was the best solution for the cheese producer, but the concern about the contamination was very important.

- A separator was installed in the incoming steam supply line, which removes a high percentage of the entrained moisture. A fine mesh screen strainer was installed to remove solid particulate matter.
- A pneumatically actuated two port valve was installed to control tank temperature. The unit throttles the flow of steam to the tank based on the signal being transmitted by the temperature controller.
- Having removed the entrained moisture and majority of particulate matter from the steam supply, a cleanable CSF16 stainless steel steam filter was installed which is capable of removing finer particles smaller than 5 microns in size.
- A vacuum breaker was added to the system in order to prevent any of the heated water being drawn back up into the filter during certain periods of operation.
- A stainless steel injector system was installed which is capable of efficiently mixing large volumes of high pressure steam with the tank contents with little noise or tank vibration. (The customer stipulated the reduction of noise levels in the production facility.)

Benefits:

- Guaranteed steam purity and assured compliance with the 3-A Industry Standard
- Inexpensive installation compared with alternative heat exchanger packages available
- Cleanable filter element for reduced operating costs (replacement element and labor costs).
- Accurate temperature control using components of the existing system
- Quiet and efficient mixing of the steam and the tank contents
- Product contamination is minimized, the cost of which could be many thousands of dollars, loss of production or even consumer dissatisfaction.

Temperature Control Valves for Liquid Service

Temperature control valves for liquid service can be divided into two groups. Normally associated with cooling, these valves can also be used on hot water.

Direct Operated Valves

Three types are available for liquid service and a selection would be made from one of the following styles.

2-Port Direct-Acting.
Normally open valve that the thermal system will close on rising temperature and used primarily for heating applications.

2-Port Reverse-Acting.
Normally closed valve which is opened on rising temperature. For use as a cooling control, valve should contain a continuous bypass bleed to prevent stagnate flow at sensor.

3-Port Piston-Balanced.
This valve is piped either for hot/cold mixing or for diverting flow between two branch lines.

Pneumatically Operated Valves

As with direct operated valves, the pneumatically operated types have the same three groups. The major difference is they require an external pneumatic or electric (through a positioner or converter) signal from a controller.

Heating And Cooling Loads

Formulas for calculating the heating or cooling load in gallons per minute of water are:

Heating Applications:

a. Heating water with water

Heating water GPM required

$$= \text{GPM (Load)} \times \frac{TR}{\Delta T_1}$$

b. Heating oil with water

Heating water GPM required

$$= \text{GPM (Load)} \times \frac{TR}{2 \times \Delta T_1}$$

c. Heating air with water

Heating water GPM required

$$= \frac{\text{CFM} \times TR}{400 \times \Delta T_1}$$

Cooling Applications:

d. Cooling air compressor jacket with water

Cooling Water GPM required

$$= \frac{42.5 \times \text{HP per Cylinder}}{8.3 \times \Delta T_2}$$

Where:

GPM = Gallons per minute water

TR = Temperature rise of heated fluid, °F

CFM = Cubic feet per minute Air

ΔT_1 = Temperature drop of heating water, °F

ΔT_2 = Temperature rise of cooling water, °F

Water Valve Sizing

Water valve capacity is directly related to the square root of the pressure drop across it, not the static system pressure. Knowing the load in GPM water or any other liquid, the minimum valve Cv required is calculated from the allowable pressure drop (ΔP):

$$Cv = GPM \sqrt{\frac{S.G.}{\Delta P}}$$ derived from...

$$GPM(Water) = Cv \sqrt{\Delta P}$$

(Other SG Liquids)

$$GPM = Cv \sqrt{\frac{\Delta P}{S.G.}}$$

If the allowable differential pressure is unknown, the following pressure drops may be applied:

- Heating and Cooling systems using low temperature hot water (below 212°F)—
  Size valve at 1 psi to 2-1/2 psi differential.
- Heating Systems using water above 212°F—
  Size valve on a 2-1/2 to 5 psi differential.
- Water for process systems—
  Size valve for pressure drop of 10% up to 20% of the system pressure.
- Cooling Valves—
  Size for allowable differential up to full system pressure drop when discharging to atmosphere. Be sure to check maximum allowable pressure drop of the valve selected. A bellows-balanced type may be required.

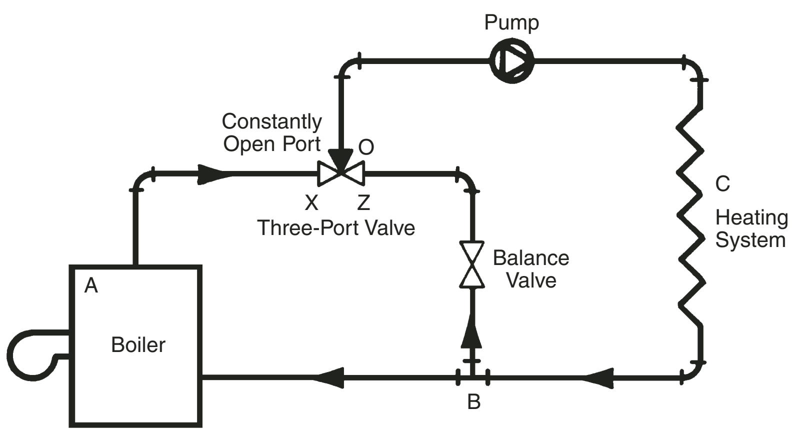


Figure 43
Three-Port Mixing Valve in a Closed Circuit
(Constant Volume, Variable Temperature)

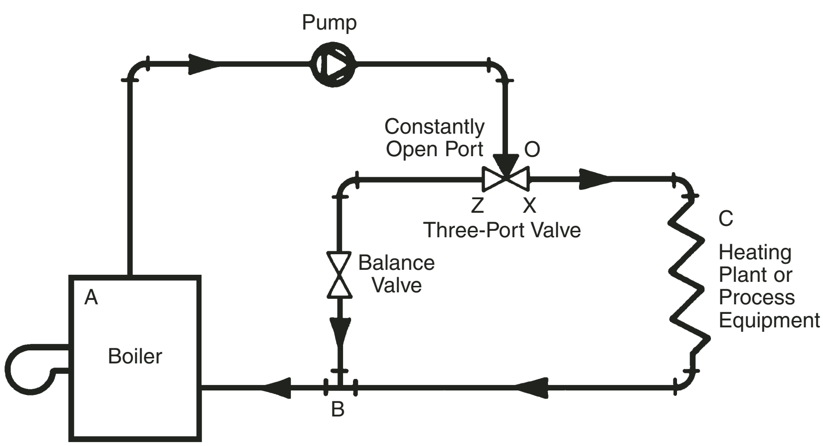


Figure 43A
Three-Port Diverting Valve in a Closed Circuit
(Constant Temperature, Variable Volume)

Using Two-Port and Three-Port Valves

Only two-port valves are used on steam systems. However, when dealing with controls for water we can select either two-port or three-port valves. But we must consider the effects of both types on the overall system dynamics.

A three-port valve, whether mixing or diverting, is fairly close to being a constant volume valve and tends to maintain constant pressure distribution in the system, irrespective of the position of the valve.

If a two-port valve were used, the flow decreases, the valve closes and the pressure or head across it would increase. This effect is inherent in the use of two-port valves and can affect the operation of other subcircuits.

Furthermore, the water standing in the mains will often cool off while the valve is closed. When the valve reopens, the water entering the heat exchanger or load is cooler than expected, and it is some time before normal heating can commence. To avoid this, a small bypass line should be installed across the supply and return mains. The bypass line should be sized to handle flow rate due to mains losses but in the absence of information, the bypass should be sized for 10% of the design flow rate.

Temperature Control Valves for Liquid Service

Mixing And Diverting Three-Port Valves

A three-port temperature control has one port that is constantly open and it is important that the circulating pump is always positioned on this side of the system. This will prevent the risk of pumping against dead end conditions and allow correct circulation to do its job.

The valve can be used either to mix or divert depending upon how it is piped into the system. A mixing valve has two inlets and one outlet, a diverting valve has one inlet and two outlets.

Fig. 43 illustrates the three-port valve used as a mixing valve in a closed circuit. It has two inlets (X and Z) and one outlet (O) which is the permanently open port. Port X is the port open on startup from cold, while Port Z will normally be closed on startup from cold. The amounts of opening in Ports X and Z will be varied to maintain a constant outlet temperature from Port O. Thus a certain percentage of hot boiler flow water will enter through Port X to mix with a corresponding percentage of cooler return water via Port Z.

When the three-port valve is used to blend cold supply water with hot water which may be from another source, for use in showers or similar open circuits where all the water does not recirculate, it is essential that the pressure of the supplies be equal. For these applications, it is recommended that both the X and Z ports be fitted with check valves to prevent any scalding or other harmful back-flow condition.

With the valve connected as shown in Fig. 43A, we now have a diverting arrangement. The valve has one inlet and two outlets. Hot water enters Port O and is either allowed through Port X to the equipment or through Port Z to return to the boiler.

The Need for Balancing

The action of a three-port valve in a closed circuit system, whether mixing or diverting, tends to change the pressure conditions around the system much less than does a two-port valve. This stability is increased greatly when a balancing valve is fitted in the bypass (or mixing connection) line. Not fitting a flow balancing valve may result in short circuiting and starvation of other subcircuits.

The balancing valve is set so that the resistance to flow in the bypass line equals or exceeds that in the load part of the subcircuit.

In Fig. 43, the balance valve must be set so that the resistance to flow in line B-Z is equal to the resistance to flow in line B-A-X. In Fig. 43A, resistance B-Z must equal resistance X-C-B.

Makeup Air Heating Coils

Air heating coils in vented condensate return systems, especially preheat coils supplied with low pressure steam modulated by a control valve, can present difficulties in achieving satisfactory drainage of condensate. There is no problem at full load with properly designed equipment, but part load conditions often lead to flooding of the coils with condensate, followed by waterhammer, corrosion and sometimes by freeze-up damage. These problems are so widespread that it is worth examining their causes and remedies in some detail.

Coil Configurations

The coils themselves are usually built with a steam header and a condensate header joined by finned tubes. The headers may be both at one side of the unit, with hairpin or U tubes between them, or sometimes an internal steam tube is used to carry the steam to the remote end of an outer finned tube. Vertical headers may be used with horizontal finned tubes,

Case in Action: Hydrogen Compressor Cooling Jacket Temperature Control

Hydrogen gas is an important ingredient to many oil refining processes. Large multi-stage compressors are located in operating sections throughout the refinery. Considerable attention is paid to maintaining gas quality, and keeping liquid from accumulating in the system.

The telltale signs of entrained liquid became evident as a high-pitched whistling noise was heard coming from the compressor sections. It was determined to be the result of poor cooling water temperature control. The cooling water/Glycol mixture leaving the heat exchanger at 95°F, circulating through the compressor jacket was causing excess hydrogen condensing on the cold surfaces of jacket walls. It's important to maintain the 95°F heat exchanger outlet temperature to assure that sufficiently-cool water/Glycol is supplied to the compressor sections necessary for proper heat transfer.

Solution

A 2" temperature control with adjustable bleed and a sensing system was installed on the cooling water/Glycol outlet piping from three stages for each of two compressors. They were set to maintain a discharge temperature of 140°F. This had the effect of holding back Glycol in the jacket sufficiently to prevent excess hydrogen condensing while, at the same time, maintaining necessary cooling.

Benefits

- Reduced energy consumption as hydrogen condensing is reduced.
- Installation of a self-contained control was far less expensive than a more sophisticated pneumatic type that was also under consideration.
- System start-up was fast because of the easily-adjusted, pre-calibrated sensing system.
- Accurate process temperature control of each jacket resulted from having separate controls on each.

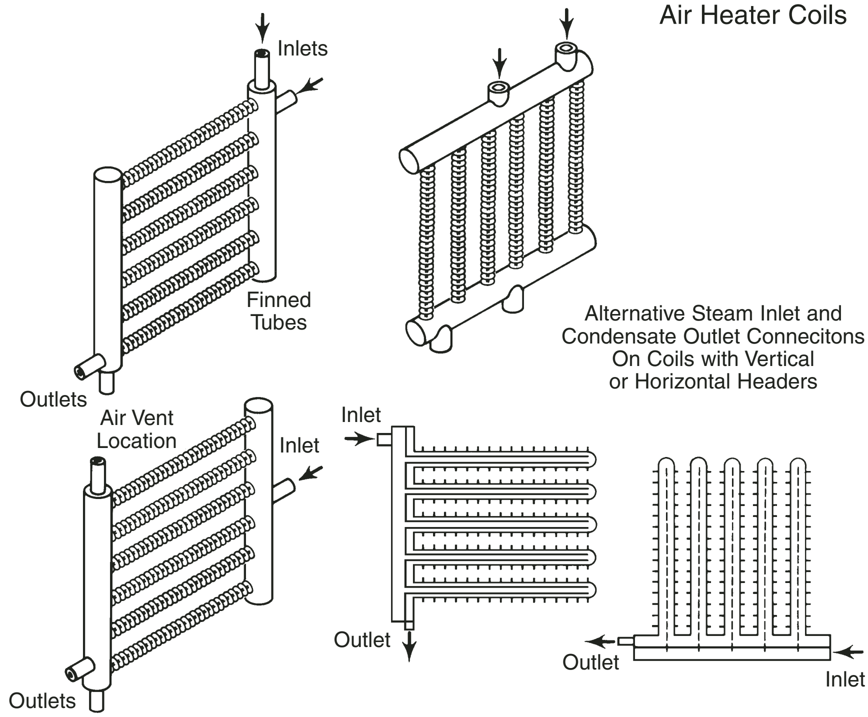
or sometimes horizontal headers at the bottom of the unit supply vertical finned tubes. The alternative arrangement has the headers at opposite sides of the unit, either horizontally at top and bottom or vertically at each side.

While each different arrangement has its own proponents, some general statements can be made, including the fact that even so-called "freeze-proof" coils can freeze if not properly drained of condensate. In "horizontal" coils, the tubes should not be horizontal but should have a slight fall from inlet to outlet so that condensate does not collect in pools but drains naturally. Steam inlets to "horizontal" headers may be at one end or at mid length, but with vertical headers the steam inlet is preferably near the top.

Venting Air From Coils

As steam enters a coil it drives air ahead of it to the drain point, or to a remote area furthest from the inlet. Coil size and shape may prevent a good deal of air from reaching the trap and as steam condenses, a film of air remains reducing heat transfer. Coils with a center inlet connection make it more difficult to ensure that air is pushed from the top tubes, the steam tending to short circuit past these tubes to the condensate header. Automatic air venting of the top condensate header of these coils is essential. With other layouts, an assessment must be made of the most likely part of the unit in which air and noncondensable gases will collect. If this is at the natural condensate drain point, then the trap must have superior air venting capability and a Float-Thermostatic type is the first choice. When an inverted bucket or other type with limited air capacity is used, an auxiliary air vent should be piped in parallel above the trap. As a general rule, a thermostatic vent and vacuum breaker are desirable on most coils to prevent problems.

Figure 44
Air Heater Coils



Waterlogged Coils

The most common cause of problems, however, is lack of pressure within the steam space under part load conditions to push condensate through the traps, especially if it is then to be lifted to a return line at high level or against a back pressure. System steam pressure lifts condensate, not the trap, and is generally not appreciated how quickly the pressure within the steam space can be reduced by the action of the control valve. When pressure used to push condensate through the traps is lost, the system "stalls" and as condensate backs up into the coil, waterlogging problems of hammering, temperature stratification, corrosion and freeze-up begin. The coil must be fitted with a vacuum breaker so that condensate is able to drain freely to the trap as shown in Fig. II-27 (page 97) and from the trap by gravity to a vented receiver and return pump. This is especially important when incoming air temperature can fall below freezing. With low coils, this may require the pump to be placed in a pit or lower floor. How to determine "system stall" conditions and the solution for draining coils to a pressurized return is covered later in this manual.

Vacuum Breaker And Trap Location

A vacuum breaker ensures that some differential pressure can always exist across a trap that drains by gravity but any elevation of condensate after the trap reduces the hydraulic head available. Heating is done using an atmospheric air/steam mixture so coil air venting is most important. A vacuum breaker should be fitted to the steam supply pipe, between the temperature control valve and the coil inlet. It is not recommended to fit a vacuum breaker on the steam trap where the hydraulic head of water used to push condensate through the trap would hold the vacuum breaker closed.

In systems where the return piping is kept under vacuum, a reversed swing check valve should be used and piped to equalize any coil vacuum not to atmosphere, but to the discharge side of the trap.

Makeup Air Heating Coils

The steam trap must handle lots of air and drain condensate at saturated steam temperature continuously while the load and pressure are changing and thus a Float-Thermostatic type is recommended for all air heating coils. The trap is mounted below the condensate outlet from the coil with a vertical drop giving enough hydraulic head to enable a suitable size to be selected. A 14" head should be the minimum and represents about 1/2 psi, a 28" head about 1 psi, and to reduce possibility of freeze-up, a drop of 3 ft. to the trap is recommended.

Preheat/Reheat Coils

The preheat/reheat coil hookup shown in Fig. II-26 (page 96) may employ a direct-acting temperature control or with larger coils, a quicker responding pilot-operated type with a closer control band is recommended. This arrangement allows filtration and perhaps humidification of the air to be carried out at the controlled preheat temperature, and the reheat coil brings the dry bulb temperature of the conditioned air to the required value for distribution. The preheat coil is used to heat outside air up to the intermediate temperature but as outside temperature increases, the temperature control lowers the steam pressure in the preheat coil and condensate drainage tends to slow down. If the coil is being used where design loads occur at subzero temperatures, there can sometimes be only atmospheric pressure in the coil, although the air passing over it is still cold enough to lead to freeze-up problems.

This difficulty is greatly reduced if the temperature sensor controlling the steam supply to the preheat coil is set to the needed distribution temperature. Part load conditions would then lead firstly to lowering the steam pressure in the reheat coil, where freezing will not occur, but pressure is maintained in the preheat coil until outside air temperatures are above the danger point. Such an arrangement reduces freeze-up problems in many instances on existing installations, at minimal cost.

Corrosion And Waterhammer Problems

Condensate mixed with air becomes corrosive and assuming the boiler water treatment is satisfactory, coil corrosion problems are usually due to condensate regularly backing up or lying stagnant on the bottom of the tubes during shutdown. If the coil is trapped correctly, the most likely cause is an overhead return which prevents the coil from draining. One remedy for this is to fit a liquid expansion steam trap at the lowest piping level, as shown in Fig. II-26 (page 96), set to open when the temperature drops below 90°F. The coil then drains only cold condensate to a sewer.

In high pressure systems where waterhammer on startup remains troublesome, a “safety drain” trap is sometimes used. This consists of a stock 15 psi rated inverted bucket trap fitted above the main trap which discharges to drain whenever coil pressure is low, but due to its design locks shut at higher pressure. While this is useful on pressurized mains, the safety trap may require a pressure considerably higher than its nominal rating to lock shut and on modulating service a considerable amount of condensate may be wasted. This makes the combination pump/trap a more viable solution to this problem.

Case in Action: Air Handling System Steam Coil Drainage

Typical storage buildings are extremely large and difficult to heat. This example in specific has three floors with approximately 486,000 ft^2 of floor space and heated with 150 air handling units. These units are comprised of bay heaters, overhead door heaters and administrative office area heaters. The minimum steam supply pressure to all of them is 20 psig and are pneumatically controlled.

In the preceding 12 month period, $201,000 was spent on labor and materials to repair damaged coils. The common problem was condensate standing in the coils, unable to drain, causing erosion due to presence of carbonic acid and bulging/splitting as a result of freezing.

Solution

Starting with a training session at the facility that addressed this problem and typical solutions, Spirax Sarco’s local sales office implemented a “Cooperative Research and Development Agreement” (CRDA). The purpose of the agreement was to test a proposed solution including Pressure Powered Pumps™ and Pump/Trap combinations to eliminate system stall, thereby assuring thorough condensate drainage, regardless of supply air temperature, control valve turn-down or over-sized heaters.

A test was conducted on four air handling units. One unit was hooked up as usual, without Pressure Powered Pump™ drainage systems. The other three were drained by either open or closed loop PPP systems. Four days into the test and the unit without a PPP drainage system had three frozen coils. It was found that as outside supply air temperature dropped below 36˚F, it was necessary to close outside dampers and use 100% recirculated air, or the coils would freeze. The three units drained by PPP systems continued operating trouble-free.

Benefits

Employee Safety

- Improved indoor air quality through the use of a higher percentage of outside air supply.
- Reduced chance of injury by eliminating water leakage on the floor from broken coils and subsequent slippage.
- Fewer burns because there are fewer steam leaks.
- Greater employee awareness of hazards because of training.

Cost Savings

- Reduced steam and condensate losses resulting in energy savings.
- Reduced cost for management support (paper-work).
- Cost savings of up to 30% above the initial installation cost in a 12 month period.

Makeup air heating coils and other heat exchange equipment where the steam supply pressure is modulated to hold a desired outflow temperature must always be kept drained of condensate. Fitting a vacuum breaker and steam trap, no matter what the size, does not always result in trouble-free operation and problems with noisy, hammering, corroded and especially frozen coils are well documented. These problems are the result of coil flooding at some point when either:

a. Incoming makeup air increases above minimum design temperature, or

b. Flow rate through an exchanger decreases below the maximum equipment output.

In a steam system, temperature regulation actually means controlling the pressure. Under partial load conditions, the steam controller, whether self-acting, pneumatic or any other type, reduces the pressure until the necessary trap differential is eliminated, the system "stalls," and steam coils become waterfilled coils.

Conditions Creating "System Stall"

With the steam equipment and the operating pressure selected, the load at which any system stalls is a function of how close the equipment is sized to the actual load and any condensate elevation or other back pressure the trap is subject to.

Other less obvious things can also seriously contribute to "system stall"; for instance, overly generous fouling factors and equipment oversizing. As an example, a fouling factor of "only" .001 can result in a coil surface area increase of 50% (See Table 10). Equipment oversizing causes the system to stall faster. This is particularly the case when the heating equipment is expected to run considerably below "design load."

Saturated steam temperature is directly related to its pressure and for any load requirement, the control valve output is determined by the basic heat transfer equation, Q = UA x ΔT. With "UA" for a steam-filled coil a constant, the amount of heat supplied, "Q", is regulated by the "ΔT," the log mean temperature difference (LMTD) between the heated air or liquid and saturated steam temperature at the pressure delivered by the valve. Thus, the steam pressure available to operate the trap is not constant but varies with the demand for heat from almost line pressure down through sub-atmospheric, to complete shutdown when no heat is required. Actual differential across the trap is further reduced when the heating surface is oversized or the trap must discharge against a back pressure. Knowing these conditions, the system must be designed accordingly.

Plotting A "Stall Chart"

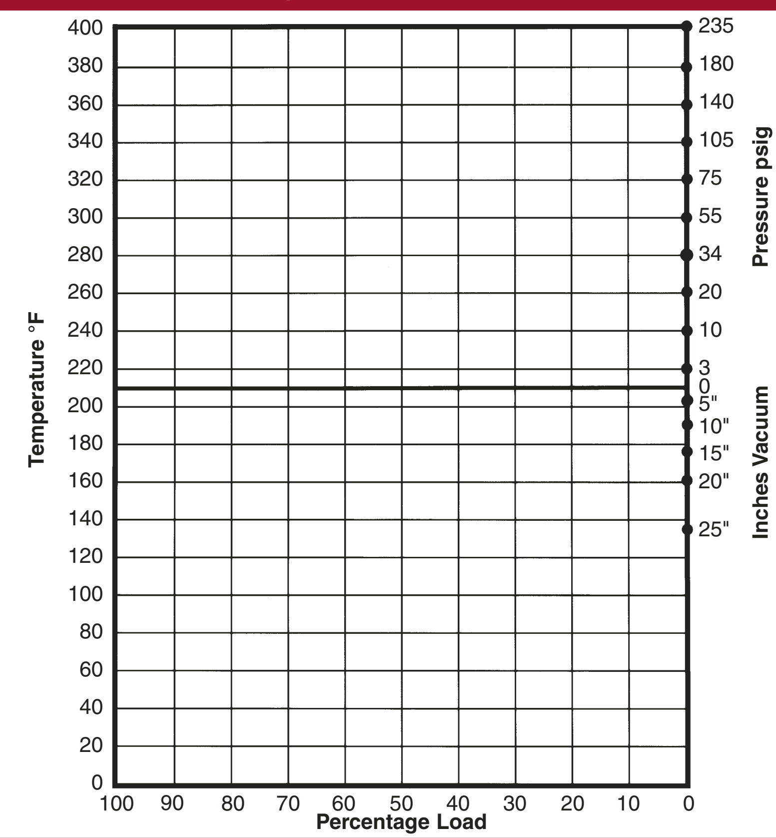
An easy way to determine the conditions at which drainage problems will occur, and prevent them at the design stage is to use the "stall chart" shown in Fig. 45.

The steam supply pressure is shown on the vertical axis, with corresponding temperatures on the opposite side, and the plot will indicate graphically what will occur for any percentage of the design load. This method provides a fairly accurate prediction of stall conditions even though the chart uses "arithmetic" rather than "log mean" temperature difference.

Table 10: Percentage Fouling Allowance

Velocity	Fouling Factor	
in Ft./Sec.	.0005	.001
1	1.14 (14%)	1.27 (27%)
2	1.19 (19%)	1.38 (38%)
3	1.24 (24%)	1.45 (45%)
4	1.27 (27%)	1.51 (51%)
5	1.29 (29%)	1.55 (55%)
6	1.30 (30%)	1.60 (60%)
7	1.31 (31%)	1.63 (63%)

Figure 45: Stall Chart



Draining Temperature Controlled Steam Equipment

An example plot is shown on Fig. 46 for a coil where air is heated to 80°F and the trap must discharge against back pressure.

Step 1. The system is designed for 100% load when air enters at 0°F (T_1) and there is 0% load when air enters at 80°F (T_2). Draw line (T_1/T2) connecting these points.

Step 2. At maximum load, the arithmetic mean air temperature (MT) is 40°F. Locate (MT) on line (T_1/T_2), extend horizontally to 0% load, and identify as (MT_1).

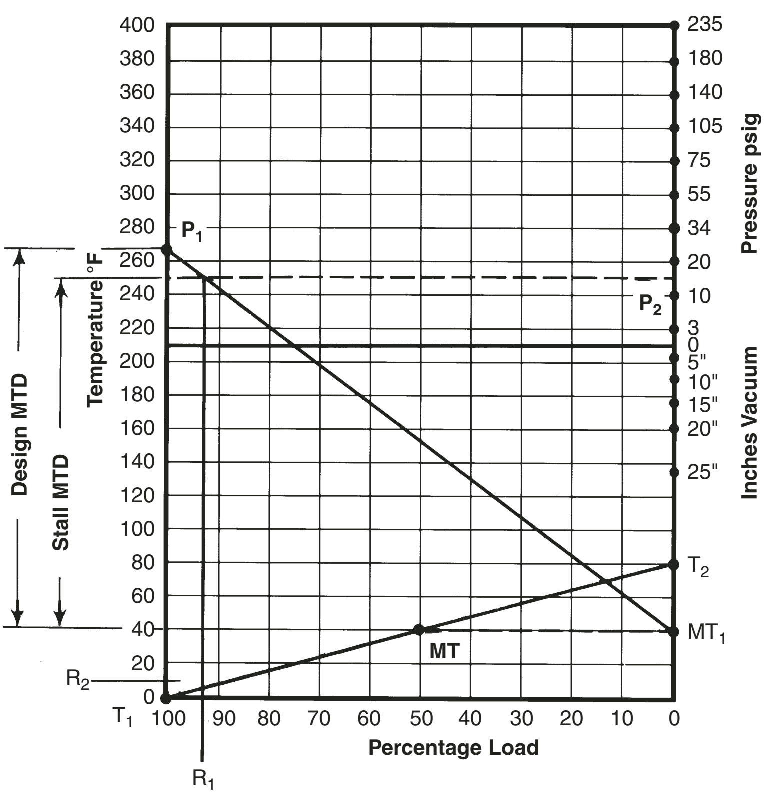
Step 3. Allowing for pressure drop, the control valve has been sized to supply 25 psig steam to the coil at 100% load. This pressure is (P_1) and has a steam temperature of 267°F. Mark (P_1) and draw line (P_1/MT_1).

Line (P_1/MT_1) approximates the steam supply at any load condition and the coil pressure is below atmospheric when it drops below the heavy line at 212°F. In a gravity system with sub-atmospheric conditions, a vacuum breaker and hydraulic pressure due to condensate will prevent stall and allow the trap to drain the coil.

Step 4. In many systems, the trap does not discharge freely to atmosphere and in our example, total back pressure on the trap is 15 psig, drawn as horizontal dotted line (P_2). Coil pressure equals back pressure at the intersection of (P_2) with (P_1/MT_1) which when dropped vertically downward to (R_1) occurs at 93% load. At less than this load, the required trap differential is eliminated, the system "stalls," and the coil begins to waterlog. In our air heating coil the air flows at a constant rate and extending the air temperature intersection horizontally to (R_2), stall occurs when the incoming air is 6°F or more.

The same procedure applies to a heat exchanger although the example temperature is not a common one. If the stall chart example represented a heat exchanger where the liquid was to be heated through a constant temperature rise from 0 to 80°F, but at a flow rate that varies, stall would still occur below 93% load. In this instance, if 100% load represents a 50 GPM exchanger, the system would stall when the demand was 46.5 GPM (50 x .93) or less.

Figure 46: Air Make-up Coil Stall Chart



Draining Equipment Under "Stall" Conditions

"System stall" is lack of positive differential across the steam trap and temperature controlled equipment will always be subject to this problem when the trap must operate against back pressure. Under these conditions, a vacuum breaker is ineffective because "stall" always occurs above atmospheric pressure. Even when steam is supplied at a constant pressure or flow to "batch" type equipment, stall can occur for some period of time on startup when the steam condenses quickly and the pressure drops below the required differential.

What happens when the system stalls is that the effective coil area ("UA" in the formula) drops as the steam chamber floods and heat transfer is reduced until the control valve responds to deliver an excessive supply of steam to the coil. This results in a "hunting system" with fluctuating temperatures and hammering coils as the relatively cooler condensate alternately backs up, then at least some portion is forced through the trap.

The solution to all system stall problems is to make condensate drain by gravity. Atmospheric systems tend to operate more predictably and are generally easier to control but major heating equipment is usually not drained into an atmospheric return because of the large amount of energy that is lost from the vent. In many process plants, venting vapors of any type is discouraged and a "closed loop" system is not only required but is less subject to oxygen corrosion problems.

Closed Loop Drainage Systems

To make equipment drain by gravity against back pressure, the steam trap must be replaced by a Pressure Powered Pump™ or pump/trap combination installed in a closed loop system. In this arrangement, the equipment does not have a vacuum breaker but is pressure equalized to drain by gravity, then isolated while condensate is pumped from the system. The basic hookup is shown in Fig. II-32 (page 99) where the equipment is constantly stalled and back pressure always exceeds the control valve supply pressure.

In many closed loop applications, the pump alone is not suitable because the steam supply pressure can at times exceed the back pressure (P_1 is higher on the "Stall Chart" than P_2.) These applications require the Pressure Powered Pump™ to be fitted in series with a Float and Thermostatic trap (combination pump-trap) to prevent steam blowthrough at loads above the stall point.

With pressurized returns and larger coils, it is often economical to fit a combination pump/trap to each coil in a closed loop system rather than the conventional gravity drain line accepting condensate from several traps and delivering it to a common pump. The pump/trap system is illustrated in Fig. II-35 (page 101) with the check valve fitted after the trap. This hookup assures maximum heat from the equipment and provides the additional advantages of no atmospheric venting, no vacuum breakers, therefore less oxygen contamination and no electric pump seals to leak. Integral to the design of this system is the air vent for startup, the liquid reservoir for accumulation during discharge, and consideration should also be given to shutdown draining with a liquid expansion steam trap.

Sizing A Combination Pump/Trap

The Pressure-Powered Pump™ selected must have capacity to handle the condensate load from the equipment at the % stall condition. Trap sizing is more critical and should be a high capacity Float and Thermostatic type sized not for the equipment load, but to handle the high flow rate during the brief pump discharge period.

The trap must be capable of handling the full system operating pressure with a capacity of stall load at 1/4 psig. This size trap will allow the pump to operate at its maximum capacity.

Multiple Parallel Coils With A Common Control Valve

While group trapping should generally be avoided, a system with a single control valve supplying steam to identical parallel coils within the same air stream can be drained to a single pump/trap combination closed loop system. (See Fig. 47.) This hookup requires that the pressure must be free to equalize into each coil. No reduced coil connections can be permitted and the common condensate manifold must not only pitch to the pump but be large enough to allow opposing flow of steam to each coil while condensate drains to the pump/trap. The basic premise still applies, that coils which are fully air vented and free to drain by gravity give maximum heat output.

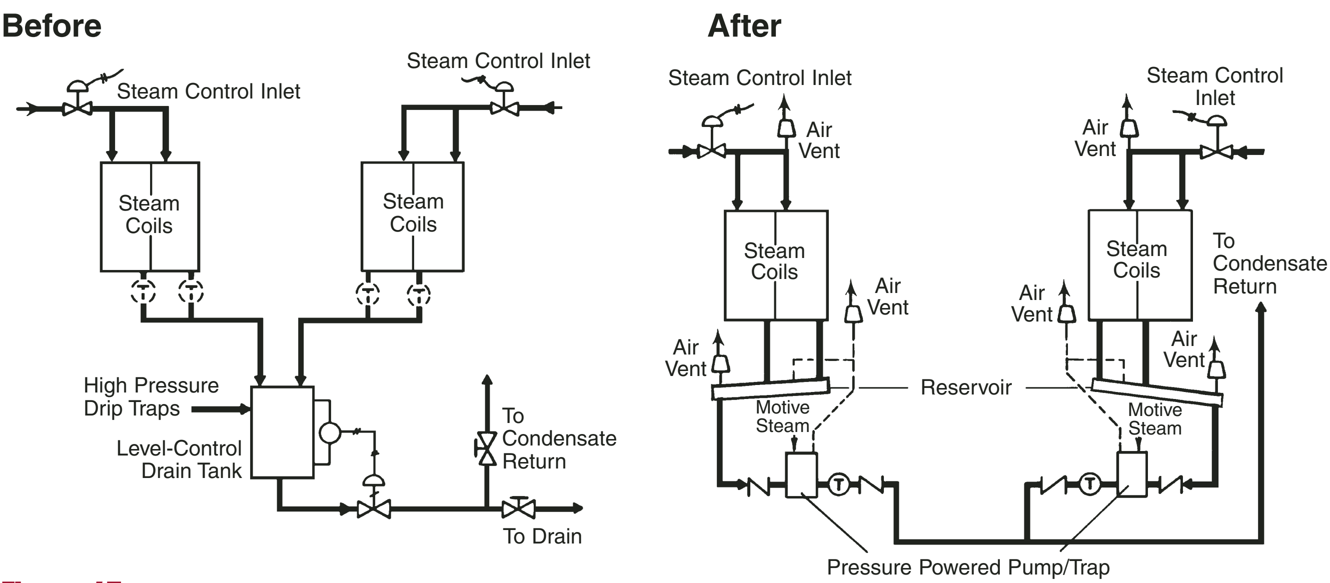


Figure 47
Combination Pressure-Powered Pump/Traps in a Closed Loop Eliminate Waterlogging in Parallel Steam Coils Previously Trapped to a "Stalled" Level Control System

Draining Temperature Controlled Steam Equipment

Case in Action: Absorption Chiller, Condensate Drainage

Absorption chillers are important sources of cooling necessary for many refinery processes. A typical example is the need to cool products (using large heat exchangers) after the stripping process in an "alky" unit. Products going to storage are generally maintained below 100°F.

Steam is used to drive the absorption process at low pressure, typically below 15 psig. Condensate drainage becomes a very real concern.

In this case, steam is supplied at 12 psig to the chiller through an automatic control valve. Condensate system backpressure is a constant 6-7 psig, considering the 30 ft. uphill pipe-run to the vented condensate receiver. The Refinery Contact Engineer recognized the potential for system stall (having previously used the Pressure Powered Pump™ to overcome other similar problems).

Solution

Two Pressure Powered Pumps™ were installed in parallel, along with necessary steam traps, air vents and strainers . The Refinery supplied the reservoir and interconnecting piping.

Benefits

- Regardless of varying steam supply pressure, considering the throttling that naturally occurs through the automatic control valve, thorough condensate drainage is assured and cooling efficiency is maintained.
- Installation cost was much lower with the Pressure Powered Pumps™ over electric pumps that were also being considered. Costly water and explosion proof control panels were not required.
- Pump maintenance cost is also much lower through elimination of the need for mechanical seals and pump motors.

Multi-Coil Heaters

In many cases, a fluid is heated by passing it through a series of heat exchangers which are all provided with steam through a common control valve (Fig. 48). Multiple section air heater coils or "batteries" typify such applications, as also the multi-roll dryers used in laundries. While the load on the first heater is usually appreciably greater than the load on later heater sections, the proportion of the total load which each section takes is often a matter of "rule of thumb" or even conjecture.

The temperature difference between the steam and the entering cold fluid can be designated Δt_1. Similarly, the temperature difference between the steam and the outlet heated fluid can be Δt_0. The ratio between Δt_1 and Δt_0 can be calculated, and will always be less than one, see Fig. 49 (page 37)

If the chart at Figure 50 is entered on the horizontal axis at this ratio, a vertical can be taken upwards until the curve corresponding with the number of heaters or coils in use is intersected. A horizontal from this point given the proportion of the total heater load which is carried by the first section.

Multiplying this proportion by the total load given the condensate rate in this section, and enables a trap with sufficient capacity to be selected.

If it is required to accurately determine the load in the second section, estimate the temperature at the outlet from the first section, and regard this as the inlet temperature for an assembly with one less section than before. Recalculate the ratio $t_s - t_o/t_s - ti_2$, and re-enter the chart at this value to find the proportion of the remaining load taken by the "first" of the remaining sections.

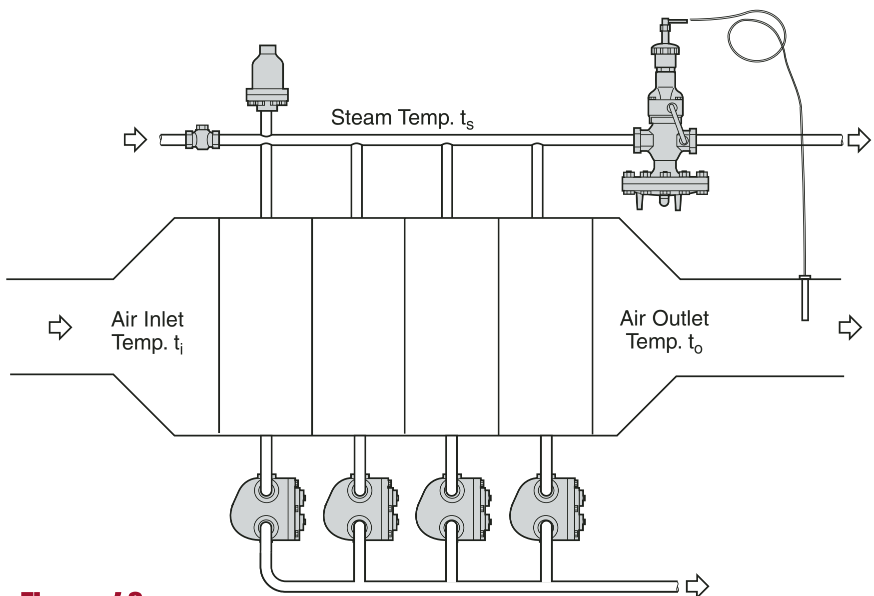


Figure 48
Multiple Coil Air Heater

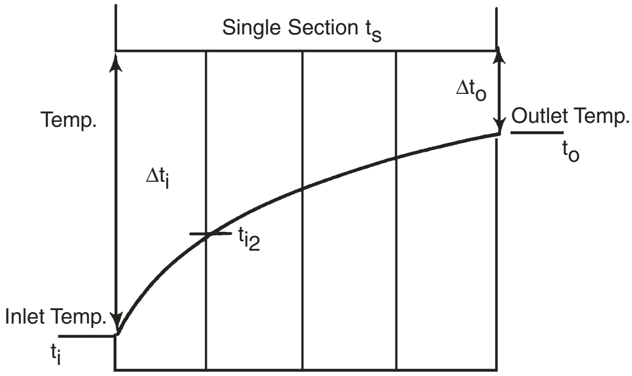


Figure 49
Temperature Distribution in Multi-Coil Heater

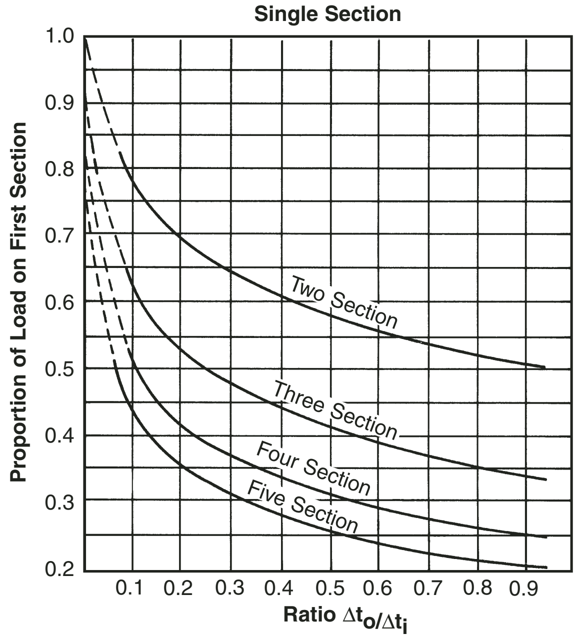


Figure 50
Load on First Section of Multi-Coil Heater

Case in Action: Air Make-Up Coil Drainage

Paper Mills require a huge volume of air exchange. This means that a great deal of air heating is necessary, particularly during winter months. Air Make-Up systems are split between two general applications:

a. Machine or Process Air Make-Up is supplied to the immediate area around the machine and, more specifically, to certain areas within the machine for higher temperature heating (i.e. Pocket Ventilation or PV coils).

b. Mill Air Make-Up, which is distributed across the mill for HVAC comfort.

Either application may be accomplished with single banks of coils or double-preheat/reheat coils, depending on heating requirements.

The mill experienced a chronic problem of frozen Air Make-Up coils, typically associated with condensate flooding and waterhammer. The 50/150 psig steam coils ballooned and ruptured routinely, creating costly maintenance headaches and safety hazards. Several coils were removed from service, requiring extensive repair.

Solution

Mill Engineers, working with the local Spirax Sarco Representative, developed a long-range plan to redesign and retrofit the entire Air Make-Up System. Over the last two years, approximately 20 Pressure Powered Pump™/float & thermostatic trap closed-loop packages have been installed. The project will continue until the entire system is retrofitted. They have similarly retrofitted several shell and tube heat exchangers, improving water heating efficiency.

Benefits

- Energy savings are achieved through installation of pressurized closed-loop packages. There is no loss to flash.
- Chemical savings are achieved because of the pressurized packages. Chemicals are not lost out the vent.
- Desired air heating efficiency has been achieved. All retrofitted coils have operated properly and continuously since installation. Flooding has been eliminated.
- Maintenance costs dropped dramatically with elimination of condensate flooding, water-hammer and freezing.
- Personnel safety has improved as steam/condensate leaks have been reduced.

Steam Trap Selection

A full discussion of steam trap functions are found in the companion Fluid System Design volume, “STEAM UTILIZATION.” The material covers operation of all types of traps, along with the need for proper air venting and trap selection. Traps are best selected not just on supply pressure and load requirements, but after reviewing the requirements of the application compared to trap characteristics including discharge temperature, air venting capability, response to pressure and load change, and resistance to dirt, corrosion, waterhammer and freezing conditions. Answering these questions leads to the selection of the most appropriate generic type of trap and the general recommendations found in Table 11 reflect this. This Selection Guide covers most trap uses and the recommended type can be expected to give satisfactory performance.

Steam Trap Sizing

Steam main drip traps shall be sized with a 2 times safety factor at full differential pressure. In most cases, they will be 3/4” size with low capacity orifice or smaller unless otherwise shown on the drawings and they shall be located every 200 feet or less. Traps for steam tracing shall be 1/4" to 1/2" size. They shall be located every 100 feet or less. Radiator traps shall be pipe size. Freeze protection traps shall be 1/2" to 3/4" size unless otherwise noted.

Traps for equipment drainage are sized with safety factors that reflect the differences of the HVAC and Process industries, such as variations in actual hydraulic head and material construction of tube bundles. A summary of these typical recommendations are as follows:

HVAC Industry

- Non-modulating control systems have traps selected with a 2 times factor at full pressure differential.
- Modulating control systems with less than 30 psig inlet pressure have traps selected for full-load at 1/2 psi pressure differential, provide 18 to 24" drip leg for condensate to drain freely to 0 psi gravity return. (With drip legs less than 18", consult a Spirax Sarco representative.)
- Modulating control systems with greater than 30 psig inlet pressure have traps selected with a 3 times factor at full pressure differential for all preheat coils, and a 2 times factor for others.

Process Industry

- Non-modulating control systems have traps selected with a 2 times factor at full pressure differential.
- Modulating controls systems with less than 30 psig inlet pressure have traps selected for full load at 1/2 psi pressure differential, provide 18 to 24" drip leg for condensate to drain freely to gravity return at 0 psi. (With drip legs less than 18", consult a Spirax Sarco representative.)
- Modulating control systems have traps selected with a 3 times factor at full pressure differential.

Case in Action: Polyvinyl Butyral Extruders

Condensate removal was needed from 3 polyvinyl butyral extruders at a pressure of 240 psi. Application required that a consistent temperature be maintained the length of the extruder to provide product quality in the melt. There were nine sections per extruder.

The customer had used various brands of traps and trap styles to drain the extruders. Most recently they used a competitors bimetallic trap. They were experiencing inconsistent temperatures throughout the length of the extruder because the bimetallic traps subcooled the condensate, which then backed up into the heat transfer area. They were also experiencing high maintenance costs in relation to these traps.

Solution

Float & Thermostatic steam traps were recommended for draining the extruders. This would give them immediate condensate removal; therefore maintaining a consistant temperature throughout the length of the extruder, providing better control over product melt. Also, upon recommendation, strainers were installed before the traps to help keep dirt out, and cut down on maintenance cost.

Benefits

- Maintained consistent temperatures with existing equipment because there is no condensate in the heat transfer area.
- There is less maintenance cost due to the strainers installed before the traps.

Steam Trap Selection Software

Selecting the best type and size steam trap is easier today for system designers who use computer software programs. The Spirax Sarco "STEAM NEEDS ANALYSIS PROGRAM" is available at www.snapfour.com and goes a step further. SNAP not only recommends and sizes the trap from input conditions, but also specifies condensate return pumps, other necessary auxiliary equipment, and warns of system problems that may be encountered. The SNAP program is user-friendly, menu-driven software that accurately calculates the condensate load for a wide range of drip, tracing and process applications (described both by common name and generic description.) Significant is the fact that a SNAP user has the choice of selecting either a recommended type of trap or a different type that may be preferred for any reason. For all selections, a formal specification sheet may be printed which contains additional information.

A QUICK GUIDE TO THE SIZING OF STEAM TRAPS

Need To Know:

1. The steam pressure at the trap—after any pressure drop through control valves or equipment.
2. THE LIFT, if any, after the trap.
   Rule of thumb: 2 ft. = 1 psi back pressure, approximately.
3. Any other possible sources of BACK PRESSURE in the condensate return system.
   e.g. A) Condensate taken to a pressurized DA. tank.
   B) Local back pressure due to discharges of numerous traps close together into small sized return.
4. QUANTITY of condensate to be handled. Obtained from A) Measurement, B) Calculation of heat load (see page 24), and C) Manufacturer's Data
5. SAFETY FACTOR—These factors depend upon particular applications, typical examples being as follows:

	General	With Temp. Control
Mains Drainage	x2	—
Storage Heaters	x2	—
Space Unit Heaters	x2	x3
Air Heating Coils	x2	x3
Submerged Coils (low level drain)	x2	—
Submerged Coils (siphon drain)	x3	—
Rotating Cylinders	x3	—
Tracing Lines	x2	—
Platen Presses	x2	—

Rule of thumb: Use factor of 2 on everything except Temperature Controlled Air Heater Coils and Converters, and Siphon applications.

How To Use

The difference between the steam pressure at the trap, and the total back pressure, including that due to any lift after the trap, is the DIFFERENTIAL PRESSURE. The quantity of condensate should be multiplied by the appropriate factor, to produce SIZING LOAD. The trap may now be selected using the DIFFERENTIAL PRESSURE and the SIZING LOAD.

Example

A trap is required to drain 22 lb/h of condensate from a 4" insulated steam main, which is supplying steam at 100 PSIG. There will be a lift after the trap of 20 ft.

Supply Pressure	= 100 psig
Lift	= 20 ft = 10 psi approx.
Therefore Differential Pressure	= 100 – 10 = 90 psi
Quantity	= 22 lb/hr
Mains Drainage Factor	= 2
Therefore Sizing Load	= 44 lb/hr

A small reduced capacity Thermo-Dynamic® steam trap will easily handle the 44 lb/h sizing load at a differential pressure of 90 psi.

Table 11: Steam Trap Selection Guide

As the USA's leading provider of steam system solutions, Spirax Sarco recognizes that no two steam trapping systems are identical. Because of the wide array of steam trap applications with inherently different characteristics, choosing the correct steam trap for optimum performance is difficult. Waterhammer, superheat, corrosive condensate, or other damaging operating characteristics dramatically affect performance of a steam trap. With over 80 years of experience in steam technology, Spirax Sarco is committed to helping it's customers design, operate and maintain an efficient steam system. You have our word on it!

	1st Choice						2nd Choice					
Application	Float & Thermostatic	Thermo-Dynamic®	Balanced Pressure	Bimetallic	Liquid Expansion	Inverted Bucket	Float & Thermostatic	Thermo-Dynamic®	Balanced Pressure	Bimetallic	Liquid Expansion	Inverted Bucket
Steam Mains to 30 psig	✓											✓
30-400 psig		✓										✓
to 600 psig		✓										✓
to 900 psig		✓										✓
to 2000 psig		✓										✓
with Superheat		✓								✓		
Separators	✓											✓
Steam Tracers Critical		✓							✓			
Non-Critical			✓					✓				
Heating Equipment												
Shell & Tube Heat Exchangers	✓											✓
Heating Coils	✓											✓
Unit Heaters	✓											✓
Plate & Frame Heat Exchangers	✓											✓
Radiators			✓									
General Process Equipment												
to 30 psig	✓											✓
to 200 psig	✓											✓
to 465 psig	✓											✓
to 600 psig						✓						
to 900 psig						✓						
to 2000 psig						✓						
Hospital Equipment												
Autoclaves	✓								✓			
Sterilizers	✓								✓			
Fuel Oil Heating												
Bulk Storage Tanks			✓				✓					
Line Heaters	✓											
Tanks & Vats												
Bulk Storage Tanks			✓				✓					
Process Vats	✓							✓				
Vulcanizers		✓					✓					
Evaporators	✓											✓
Reboilers	✓											✓
Rotating Cylinders	✓											
Freeze Protection					✓							

The Formation of Flash Steam

When hot condensate under pressure is released to a lower pressure, its temperature must very quickly drop to the boiling point for the lower pressure as shown in the steam tables. The surplus heat is utilized by the condensate as latent heat causing some of it to re-evaporate into steam. Commonly referred to as “flash steam”, it is in fact perfectly good useable steam even at low pressure.

Table 12: Percent Flash

Steam Pressure psig	Atmosphere 0	Flash Tank Pressure 2	5	10	15	20	30	40	60	80	100
5	1.7	1.0	0								
10	2.9	2.2	1.4	0							
15	4.0	3.2	2.4	1.1	0						
20	4.9	4.2	3.4	2.1	1.1	0					
30	6.5	5.8	5.0	3.8	2.6	1.7	0				
40	7.8	7.1	6.4	5.1	4.0	3.1	1.3	0			
60	10.0	9.3	8.6	7.3	6.3	5.4	3.6	2.2	0		
80	11.7	11.1	10.3	9.0	8.1	7.1	5.5	4.0	1.9	0	
100	13.3	12.6	11.8	10.6	9.7	8.8	7.0	5.7	3.5	1.7	0
125	14.8	14.2	13.4	12.2	11.3	10.3	8.6	7.4	5.2	3.4	1.8
160	16.8	16.2	15.4	14.1	13.2	12.4	10.6	9.5	7.4	5.6	4.0
200	18.6	18.0	17.3	16.1	15.2	14.3	12.8	11.5	9.3	7.5	5.9
250	20.6	20.0	19.3	18.1	17.2	16.3	14.7	13.6	11.2	9.8	8.2
300	22.7	21.8	21.1	19.9	19.0	18.2	16.7	15.4	13.4	11.8	10.1
350	24.0	23.3	22.6	21.6	20.5	19.8	18.3	17.2	15.1	13.5	11.9
400	25.3	24.7	24.0	22.9	22.0	21.1	19.7	18.5	16.5	15.0	13.4

Percent flash for various initial steam pressures and flash tank pressures.

Proportion Of Flash Steam Released

The amount of flash steam which each pound of condensate will release may be calculated readily. Subtracting the sensible heat of the condensate at the lower pressure from that of the condensate passing through the traps will give the amount of heat available from each pound to provide Latent Heat of Vaporization. Dividing this amount by the actual Latent Heat per pound at the Lower Pressure will give the proportion of the condensate which will flash off. Multiplying by the total quantity of condensate being considered gives the weight of Low Pressure Steam available.

Thus, if for example, 2000 lb/h of condensate from a source at 100 psi is flashed to 10 psi, we can say:

Sensible Heat at 100 psi	=	309 Btu/lb
Sensible Heat at 10 psi	=	208 Btu/lb
Heat Available for Flashing	=	101 Btu/lb
Latent Heat at 10 psi	=	952 Btu/lb
Proportion Evaporated	=	101 ÷ 952 = 0.106 or 10.6%
Flash Steam Available	=	0.106 x 2000 lb/h
	=	212 lb/h

To simplify this procedure we can use Table 12 to read off the percentage of flash steam produced by this pressure drop. An example would be if we had 100 PSIG saturated steam/condensate being discharged from a steam trap to an atmospheric, gravity flow condensate return system (0 psig), the flash percentage of the condensate would be 13.3% of the volume discharged.

Conversely, if we had 15 psig saturated steam discharging to the same (0 psig) atmospheric gravity flow return system, the percentage of flash steam would be only 4% by volume. These examples clearly show that the amount of flash released depends upon the difference between the pressures upstream and downstream of the trap and the corresponding temperatures of those pressures in saturated steam. The higher the initial pressure and the lower the flash recovery pressure, the greater the quantity of flash steam produced.

It must be noted here that the chart is based upon saturated steam pressure/temperature conditions at the trap inlet, and that the condensate is discharged as rapidly as it appears at the trap. Steam traps that subcool the condensate, such as balanced pressure thermostatic and bimetallic traps, hold condensate back in the system allowing it to give up sensible heat energy and causing it to cool below the saturated steam temperature for that pressure. Under those circumstances, we must calculate from the formula above the percentage of flash steam produced, but the amount of subcooling (the condensate temperature) must be known before calculating.

Flash Steam Utilization

In an efficient and economical steam system, this so called Flash Steam will be utilized, on any load which will make use of low pressure steam. Sometimes it can be simply piped into a low pressure distribution main for general use. The ideal is to have a greater demand for Low Pressure steam, at all times, than available supply of flash steam. Only as a last resort should flash steam be vented to atmosphere and lost.

If the flash steam is to be recovered and utilized, it has to be separated from the condensate. This is best achieved by passing the mixture of flash steam and condensate through what is know as a “flash tank” or “flash vessel”. A typical arrangement is shown in Fig. II-76 (page 120). The size of the vessel has to be designed to allow for a reduced velocity so that the separation of the flash steam and condensate can be accomplished adequately, so as not to have carryover of condensate out into the flash steam recovery system. This target velocity is ten feet per second per ASHRAE standards to ensure proper separation. The condensate drops to the bottom of the flash tank where it is removed by a float and thermostatic steam trap. The flash steam outlet connection is sized so that the flash steam velocity through the outlet is approximately 50 ft./sec. The condensate inlet is also sized for 50 ft./sec. flash velocity.

A number of basic requirements and considerations have to be met before flash steam recovery is a viable and economical proposition:

1. It is first essential to have a sufficient supply of condensate, from loads at sufficiently higher pressures, to ensure that enough flash steam will be released to make recovery economically effective.

   The steam traps, and the equipment from which they are draining condensate, must be able to function satisfactorily while accepting the new back pressure applied to them by the flash recovery system. Particular care is needed when attempting to recover flash steam from condensate which is leaving equipment controlled by a modulating temperature control valve. At less than full loads, the steam space pressure will be lowered by the action of the temperature control valve. If the steam space pressure approaches or even falls below the flash steam vessel pressure, condensate drainage from the steam space becomes impractical by a steam trap alone, and the equipment becomes “stalled” and water logging will most definitely occur.
2. The second requirement is a suitable use for low pressure flash steam. Ideally, low pressure load(s) requires at all times a supply of steam which either equals or exceeds the available flash steam supply. The deficit can then be made up through a pressure reducing valve set. If the supply of flash steam exceeds the demand for it, the surplus may have to be vented to waste through a backpressure control valve (see Fig. II-77, page 120).

   Thus, it is possible to utilize the flash steam from process condensate on a space heating installation - but the savings will only be achieved during the heating season. When heating is not required, the recovery system becomes ineffective. Whenever possible, the better arrangement is to use flash steam from process condensate to supply process loads, and that from heating condensate to supply heating loads. Supply and demand are then more likely to remain “in step”. When all else fails, in many facilities there is always a need for hot water, especially in the boiler house. This can be supplied via a heat exchanger and the use of flash steam.
3. It is also preferable to select an application for the flash steam which is reasonably close in proximity to the high pressure condensate source. Piping for low pressure steam is inevitably of larger diameter. This makes it somewhat costly to install. Furthermore, the heat loss from large diameter pipes reduces the benefits obtained from flash steam recovery and in the worst cases could outweigh them.

Flash steam recovery is simplest when being recovered from a single piece of equipment that condenses a large amount of steam, such as a large steam to water converter of a large air handling coil bank, but we cannot forget that flash steam recovery systems by design will apply a backpressure to the equipment being utilized as the flash steam source.

How To Size Flash Tanks And Vent Lines

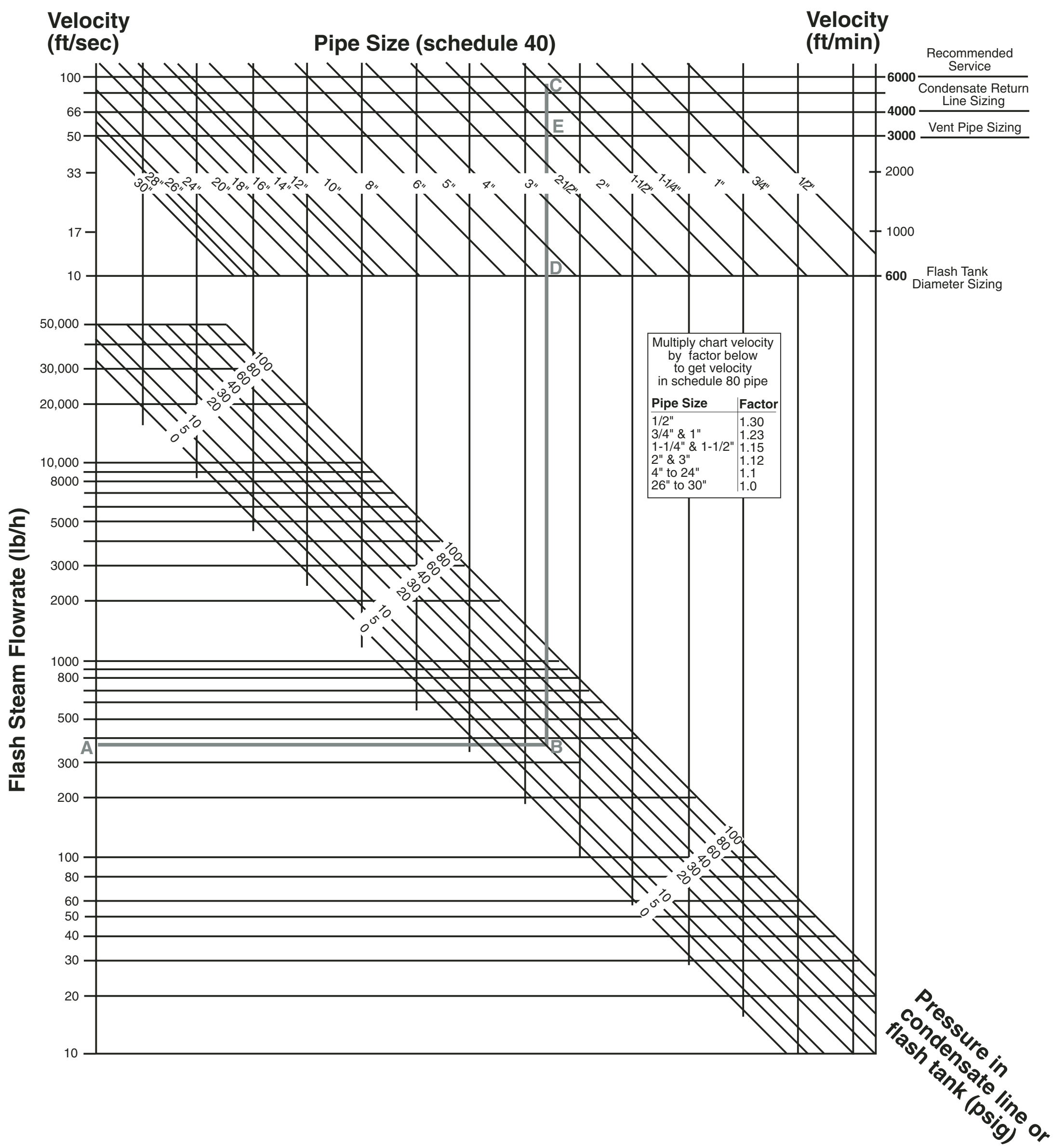
Whether a flash tank is atmospheric or pressurized for flash recovery, the procedure for determining its size is the same. The most important dimension is the diameter. It must be large enough to provide adequate separation of the flash and condensate to minimize condensate carryover.

Example

Size a 20 psig flash recovery vessel utilizing condensate from a 160 psig steam trap discharging 3000 lb/h.

1. Determine percent flash steam produced using Table 12. With a steam pressure of 160 psig and a flash tank pressure of 20 psig, read a value of 12.4%.
2. Next, multiply the condensate load by the percent flash from Step #1 to determine the flowrate, of flash steam produced. 3,000 lb/h x .124 = 372 lb/h.
3. Using the calculated flash steam quantity of 372 lb/h enter Fig. 51 at "A" and move horizontally to the right to the flash tank pressure of 20 psig "B". Rise vertically to the flash tank diameter line (600 ft/min) at "D". Read tank diameter of 5". If schedule 80 pipe is to be installed, the table within the body of the chart can be used to determine whether the velocity will exceed the recommended limit of 600 ft/min.
4. From point "D" continue to rise vertically to "E" to determine the size of vent pipe to give a velocity between 3000 and 4000 ft/min. In this case 2" schedule 40 pipe. As before, use the table within the body of chart for schedule 80 pipe.

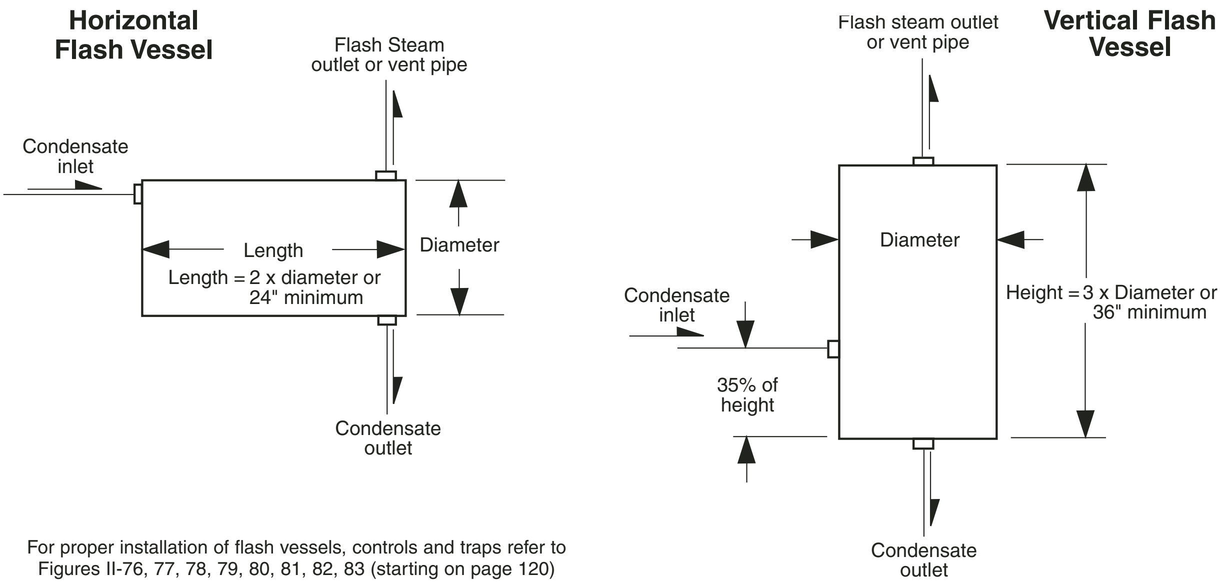
Figure 51: Condensate Line, Flash Tank, and Vent Line Sizing



Flash Vessel Configurations

Flash vessels can be either horizontal or vertical. For flash steam recovery (pressurized receiver) the vertical style is preferred because of its ability to provide better separation of steam and water.

Figure 52: Flash Vessel Configurations



For proper installation of flash vessels, controls and traps refer to Figures II-76, 77, 78, 79, 80, 81, 82, 83 (starting on page 120)

Case in Action: Sour Water Condenser-Reboiler Temperature Control

The sour water condenser-reboiler is an important element in a refinery sulfur unit. Though the configuration/design will vary with the specific process used, the purpose and priority remain the same. Process water contaminated with ammonia and hydrogen sulfide gas (H_2S) is stripped of those compounds for reuse. The remaining stream of contaminants goes to waste treatment. The process depends on accurate temperature control of the steam heated condenser-reboiler.

40 psig steam is supplied through a modulating control valve. Condensate is lifted 15 feet from the outlet at the bottom of the vertical condenser-reboiler to the overhead condensate receiver tank, which is maintained at 20 psig. Process load fluctuations and resultant turndown on the modulating steam control valve would have created a STALL situation, unacceptable process temperature control and reduced throughput.

Solution

A 3" x 2" PPF with 2-1/2" FTB 125 pump/trap combination was designed into the new project as was a VS 204 air vent. The installation was immediately successful.

Benefits

- With faster start-up, it came up to temperature faster than any other comparable unit, to date, at the refinery. This improves productivity.
- The feed rate is higher than designed because the unit is able to operate efficiently at any degree of turndown.
- Installation cost is several times less costly for the pump/trap combo than traditional level control system that would otherwise have been used.
- Maintenance cost is lower, through elimination of electric/pneumatic controls and electric pumps used in a traditional level control system.

The importance of effective condensate removal from steam spaces has been stressed throughout this course. If maximum steam system efficiency is to be achieved, the best type of steam trap must be fitted in the most suitable position for the application in question, the flash steam should be utilized, and the maximum amount of condensate should be recovered.

There are a number of reasons why condensate should not be allowed to discharge to drain. The most important consideration is the valuable heat which it contains even after flash steam has been recovered. It is possible to use condensate as hot process water but the best arrangement is to return it to the boiler house, where it can be re-used as boiler feed water without further treatment, saving preheating fuel, raw water and the chemicals needed for boiler feed treatment. These savings will be even greater in cases where effluent charges have to be paid for the discharge of valuable hot condensate down the drain.

Condensate recovery savings can add up to 20 to 25% of the plant's steam generating costs. One justifiable reason for not returning condensate is the risk of contamination. Perforated coils in process vessels and heat exchangers do exist and the cross contamination of condensate and process fluids is always a danger. If there is any possibility that the condensate is contaminated, it must not be returned to the boiler. These problems have been lessened by the application of sensing systems monitoring the quality of condensate in different holding areas of a plant to determine condensate quality and providing a means to re-route the condensate if contaminated.

Condensate Line Sizing

Condensate recovery systems divide naturally into three sections, each section requiring different design considerations.

a. Drain Lines to the traps carry pressurized high temperature hot water that moves by gravity.
b. Trap discharge lines that carry a two-phase mixture of flash steam and condensate.
c. Pumped return systems utilizing electric or non-electric pumps.

Drain Lines To Traps

In the first section, the condensate has to flow from the condensing surface to the steam trap. In most cases this means that gravity is relied on to induce flow, since the heat exchanger steam space and the traps are at the same pressure. The lines between the drainage points and the traps can be laid with a slight fall, say 1" in 10 feet, and Table 13 shows the water carrying capacities of the pipes with such a gradient. It is important to allow for the passage of incondensibles to the trap, and for the extra water to be carried at cold starts. In most cases, it is sufficient to size these pipes on twice the full running load.

Trap Discharge Lines

At the outlet of steam traps, the condensate return lines must carry condensate, non-condensible gases and flash steam released from the condensate. Where possible, these lines should drain by gravity to the condensate receiver, whether this be a flash recovery vessel or the vented receiver of a pump. When sizing return lines, two important practical points must be considered.

Table 13: Condensate, lb/h

Steel Pipe Size	Approximate Frictional Resistance in inches Wg per 100 ft of Travel			
	1	5	7	10
1/2"	100	240	290	350
3/4"	230	560	680	820
1"	440	1070	1200	1550
1 1/4"	950	2300	2700	3300
1 1/2"	1400	3500	4200	5000
2"	2800	6800	8100	9900
2 1/2"	5700	13800	16500	20000
3"	9000	21500	25800	31000
4"	18600	44000	52000	63400

First, one pound of steam has a specific volume of 26.8 cubic feet at atmospheric pressure. It also contains 970 BTU's of latent heat energy. This means that if a trap discharges 100 pounds per hour of condensate from 100 psig to atmosphere, the weight of flash steam released will be 13.3 pounds per hour, having a total volume of 356.4 cubic feet. It will also have 12,901 BTU's of latent heat energy. This will appear to be a very large quantity of steam and may well lead to the erroneous conclusion that the trap is passing live steam (failed open).

Another factor to be considered is that we have just released 13.3 pounds of water to the atmosphere that should have gone back to the boiler house for recycling as boiler feed water. Since we just wasted it, we now have to supply 13.3 pounds of fresh city water that has been softened, chemically treated and preheated to the feedwater system's temperature before putting this new water back into the boiler.

Secondly, the actual formation of flash steam takes place within and downstream of the steam trap orifice where pressure drop occurs. From this point onward, the condensate return system must be capable of carrying this flash steam, as well as condensate. Unfortunately, in the past, condensate return lines

have been sized using water volume only and did not include the flash steam volume that is present.

The specific volume of water at 0 psig is .017 cubic feet per pound, compared to 26.8 cubic feet per pound for flash steam at the same pressure. Sizing of condensate return lines from trap discharges based totally on water is a gross error and causes lines to be drastically undersized for the flash steam. This causes condensate lines to become pressurized, not atmospheric, which in turn causes a backpressure to be applied to the trap's discharge which can cause equipment failure and flooding.

This undersizing explains why the majority of 0 psi atmospheric condensate return systems in the United States do not operate at 0 psig. To take this thought one step further for those people who perform temperature tests on steam traps to determine if the trap has failed, the instant we cause a positive pressure to develop in the condensate return system by flash steam, the condensate return line now must follow the pressure/temperature relationship of saturated steam. So, trap testing by temperature identifies only that we have a return system at a certain temperature above 212°F (0 psig) and we can then determine by that temperature the system pressure at which it is operating. Elevated condensate return temperatures do not necessarily mean a trap has failed.

When sizing condensate return lines, the volume of the flash steam must be given due consideration. The chart at Fig. 51 (page 43) allows the lines to be sized as flash steam lines—since the volume of the condensate is so much less than that of the steam released.

Draining condensate from traps serving loads at differing pressures to a common condensate return line is a concept which many find difficult. It is often assumed that the HP "high pressure" condensate will prevent the "low pressure" condensate from passing through the LP traps and give rise to waterlogging of the LP system.

However, the terms HP and LP can only apply to the conditions on the upstream side of the seats in the traps. At the downstream or outlet side of the traps, the pressure must be the common pressure in the return line. This return line pressure will be the sum of at least three components:

1. The pressure at the end of the return line, either atmospheric or of the vessel into which the line discharges.
2. The hydrostatic head needed to lift the condensate up any risers in the line.
3. The pressure drop needed to carry the condensate and any flash steam along the line.

Item 3 is the only one likely to give rise to any problems if condensate from sources at different pressures enters a common line. The return should be sufficiently large to carry all the liquid condensate and the varying amounts of flash steam associated with it, without requiring excessive line velocity and excessive pressure drop. If this is accepted, the total return line cross sectional area will be the same, whether a single line is used, or if two or more lines are fitted, with each taking the condensate from a single pressure source.

The return could become undersized, requiring a high pressure at the trap discharges and restricting or preventing discharge from the LP traps, if it is forgotten that the pipe has to carry flash steam as well as water and that flash steam is released in appreciable quantity from HP condensate.

While the percentage, by weight, of flash steam may be rather low, its overall volume in comparison to the liquid is very large. By determining the quantity of flash steam and sizing the return line for velocities between 4,000 and 6,000 ft/min, the two-phase flow within the pipe can be accommodated. The information required for sizing is the condensate load in lb/h, inlet pressure to steam trap(s) in psig and return line system pressure.

Example:

Size a condensate return line from a 160 psig steam trap discharging to 20 psig. flash tank. Load is 3,000 lb/h.

1. Determine percent flash steam produced using Table 12 (page 41). With a steam pressure of 160 psig and a flash tank pressure of 20 psig read a value of 12.4%.
2. Next, multiply the condensate load by the percent flash from step #1 to determine the flowrate, of flash steam produced.

   3,000 lb/h x .124 = 372 lb/h.
3. Enter Fig. 51 (page 43) at the flash steam flowrate of 372 lb/h at "A" and move horizontally to the right to the flash tank pressure of 20 psig "B". Rise vertically to choose a condensate return line size which will give a velocity between 4,000 and 6,000 ft/min, "C". In this example, an 1-1/2" schedule 40 pipe with a velocity of approximately 5,000 ft/min. If schedule 80 pipe is to be used, refer to table within body of chart. Multiply the velocity by the factor to determine whether the velocity is within acceptable limits.

Pumped Return Lines

Finally, the condensate is often pumped from the receiver to the boiler plant. These pumped condensate lines carry only water, and rather higher water velocities can often be used so as to minimize pipe sizes. The extra friction losses entailed must not increase back pressures to the point where the pump capacity is affected. Table 35 (page 77) can be used to help estimate the frictional resistance presented by the pipes. Commonly, velocities in pumped returns should be limited to 6-8 ft./sec.

Electric pumps are commonly installed with pumping capability of 2-1/2 or 3 times the rate at which condensate reaches the receiver. This increased instantaneous flow rate must be kept in mind when sizing the delivery lines. Similar considerations apply when steam powered pumps are used, or appropriate steps taken to help attain constant flow along as much as possible of the system.

Where long delivery lines are used, the water flowing along the pipe as the pump discharges attains a considerable momentum. At the end of the discharge cycle when the pump stops, the water tends to keep moving along the pipe and may pull air or steam into the delivery pipe through the pump outlet check valve. When this bubble of steam reaches a cooler zone and condenses, the water in the pipe is pulled back towards the pump. As the reversed flow reaches and closes the check valve, waterhammer often results. This problem is greatly reduced by adding a second check valve in the delivery line some 15 or 20 ft. from the pump. If the line lifts to a high level as soon as it leaves the pump, then adding a generously sized vacuum breaker at the top of the riser is often an extra help. However, it may be necessary to provide means of venting from the pipe at appropriate points, the air which enters through the vacuum breaker. See Figures II-71 and II-72 (page 118).

The practice of connecting additional trap discharge lines into the pumped main is to be avoided whenever possible. The flash steam which is released from this extra condensate leads to thermal shock creating a banging noise within the piping commonly associated with waterhammer. The traps should discharge into a separate gravity line which carries the condensate to the receiver of the pump. If this is impossible, a second best alternative may be to pipe the trap discharge through a sparge or diffuser inside the pumped return line.

The trap most suitable for this application would be the Float and Thermostatic type due to its continuous discharge. This is very much a compromise and will not always avoid the noise (see Fig. 53 and 53A) although it will reduce the severity.

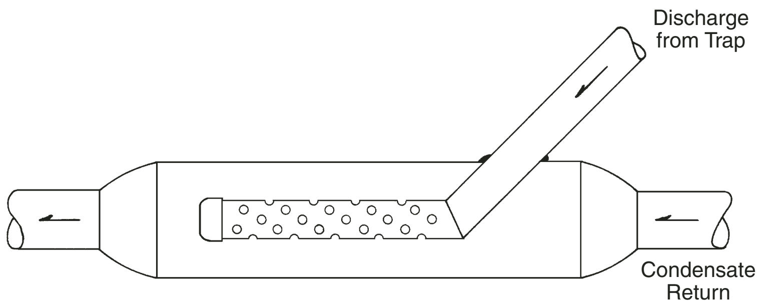


Figure 53
Discharge of Steam Trap into Pumped (flooded) Return Line using Sparge Pipe.

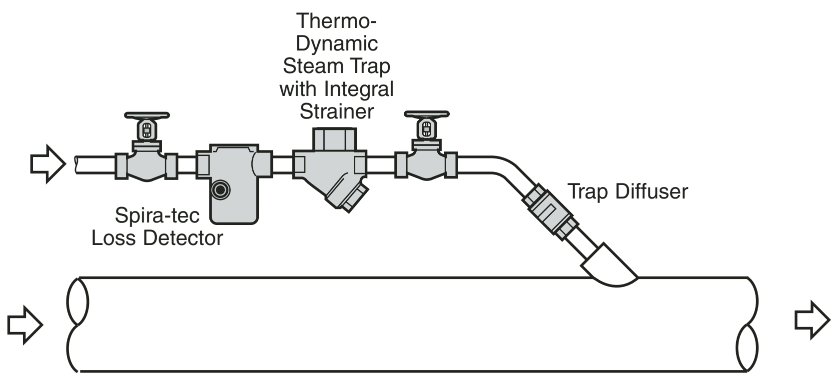


Figure 53A
Discharge of Steam Trap into Pumped (flooded) Return Line using a Trap Diffuser.

In nearly all steam-using plants, condensate must be pumped from the location where it is formed back to the boilerhouse, or in those cases where gravity drainage to the boilerhouse is practical, the condensate must be lifted into a boiler feed tank or deaerator. Even where deaerators are at low level, they usually operate at a pressure a few psi above atmospheric and again, a pump is needed to lift condensate from atmospheric pressure to deaerator tank pressure.

Electric Condensate Return Pumps

When using electric pumps to lift the condensate, packaged units comprising a receiver tank (usually vented to atmosphere) and one or more motorized pumps are commonly used. It is important with these units to make sure that the maximum condensate temperature specified by the manufacturer is not exceeded, and the pump has sufficient capacity to handle the load. Condensate temperature usually presents no problem with returns from low pressure heating systems. There, the condensate is often below 212°F as it passes through the traps, and a little further subcooling in the gravity return lines and in the pump receiver itself means that there is little difficulty in meeting the maximum temperature limitation. See Fig. II-74 (page 119).

On high pressure systems, the gravity return lines often contain condensate at just above 212°F, together with some flash steam. The cooling effect of the piping is limited to condensing a little of the flash steam, with the remainder passing through the vent at the pump receiver. The water must remain in the receiver for an appreciable time if it is to cool sufficiently, or the pump discharge may have to be throttled down to reduce the pump's capacity if cavitation is to be avoided. See Fig. II-75 (page 119).

The PUMP NPSH in any given application can readily be estimated from:

$$NPSH = hsv = \frac{144}{W}(Pa - Pvp) + hs - hf$$

Where:

- Pa = Absolute pressure in receiver supplying pump, in psi (that is at atmospheric pressure in the case of a vented receiver).
- Pvp = Absolute pressure of condensate at the liquid temperature, in psi.
- hs = Total suction head in feet. (Positive for a head above the pump or negative for a lift to the pump)
- hf = Friction loss in suction piping.
- W = Density of water in pounds per cubic foot at the appropriate temperature.

The absolute pressure at the inlet to the pump is usually the atmospheric pressure in the receiver, plus the static head from the water surface to the pump inlet, minus the friction loss through pipes, valves and fittings between the receiver and the pump. If this absolute pressure exceeds the vapor pressure of water at the temperature at which it enters the pump, then a Net Positive Suction Head exists. Providing this NPSH is above the value specified by the pump manufacturer, the water does not begin to boil as it enters the pump suction, and cavitation is avoided. If the water entering the pump is at high temperature, its vapor pressure is increased and a greater hydrostatic head over the pump suction is needed to ensure that the necessary NPSH is obtained.

If the water does begin to boil in the pump suction, the bubbles of steam are carried with the water to a high pressure zone in the pump. The bubbles then implode with hammer-like blows, eroding the pump and eventually destroying it. The phenomenon is called cavitation and is readily recognized by its typical rattle-like noise, which usually diminishes as a valve at the pump outlet is closed down.

However, since in most cases pumps are supplied coupled to receivers and the static head above the pump inlet is already fixed by the pump manufacturer, it is only necessary to ensure that the pump set has sufficient capacity at the water temperature expected at the pump. Pump manufacturers usually have a set of capacity curves for the pump when handling water at different temperatures and these should be consulted.

Where steam systems operate at higher pressures than those used in LP space heating systems, as in process work, condensate temperatures are often 212°F, or more where positive pressures exist in return lines. Electric pumps are then used only if their capacity is downrated by partial closure of a valve at the outlet; by using a receiver mounted well above the pump to ensure sufficient NPSH; or by subcooling the condensate through a heat exchanger of some type.

Pressure Powered Condensate Pump

All these difficulties are avoided by the use of non-electric condensate pumps, such as the Pressure Powered Pump™. The Pressure Powered Pump™ is essentially an alternating receiver which can be pressurized, using steam, air or other gas. The gas pressure displaces the condensate (which can be at any temperature up to and including boiling point) through a check

valve at the outlet of the pump body. At the end of the discharge stroke, an internal mechanism changes over, closing the pressurizing inlet valve and opening an exhaust valve. The pressurizing gas is then vented to atmosphere, or to the space from which the condensate is being drained. When the pressures are equalized, condensate can flow by gravity into the pump body to refill it and complete the cycle.

As the pump fills by gravity only, there can be no cavitation and this pump readily handles boiling water or other liquids compatible with its materials of construction.

The capacity of the pump depends on the filling head available, the size of the condensate connections, the pressure of the operating steam or gas, and the total head through which the condensate is lifted. This will include the net difference in elevation between the pump and the final discharge point; any pressure difference between the pump receiver and final receiver; friction in the connecting pipework, and the force necessary to accelerate the condensate from rest in the pump body up to velocity in the discharge pipe. Tables listing capacities under varying conditions are provided in the catalog bulletins.

Piping Requirements

Depending upon the application, the Pressure Powered Pump™ body is piped so that it is vented to atmosphere or, in a closed system, is pressure equalized back to the space that it drains. This allows condensate to enter the pump but during the short discharge stroke, the inlet check valve is closed and condensate accumulates in the inlet piping. To eliminate the possibility of condensate backing up into the steam space, reservoir piping must be provided above the pump with volume as specified in the catalog. A closed system requires only a liquid reservoir. In open systems, the vented receiver serves this purpose as it is always larger in order to also separate the flash steam released.

Vented Systems

Condensate from low pressure heating systems may be piped directly to a small size Pressure Powered Pump™ only when 50 lb/h or less of flash steam must vent through the pump body. This does not eliminate the requirement that there must be enough piping to store condensate during the brief discharge cycle. In many low pressure systems, the reservoir may be a section of larger horizontal pipe which is vented to eliminate flash steam. In higher pressure, high load systems, the larger quantity of flash released requires a vented receiver with piping adequate to permit complete separation. To prevent carryover of condensate from the vent line, the receiver should be sized to reduce flow velocity to about 10 FPS.

Closed Loop Systems

It is often advisable where larger condensate loads are being handled to dedicate a Pressure Powered Pump™ to drain a single piece of equipment. The pump exhaust line can then be directly connected to the steam space of a heat exchanger or, preferably with air heating coils, to the reservoir. This allows condensate to drain freely to the pump inlet and through a steam trap at the pump outlet. Only liquid is contained in the reservoir of a closed loop system. Fig.II-32 (page 99) illustrates how the Pressure-Powered pump functions as a pumping trap, and use Fig. II-35 (page 101) when the steam supply may sometimes be greater than the return pressure and a combination pump/trap is required.

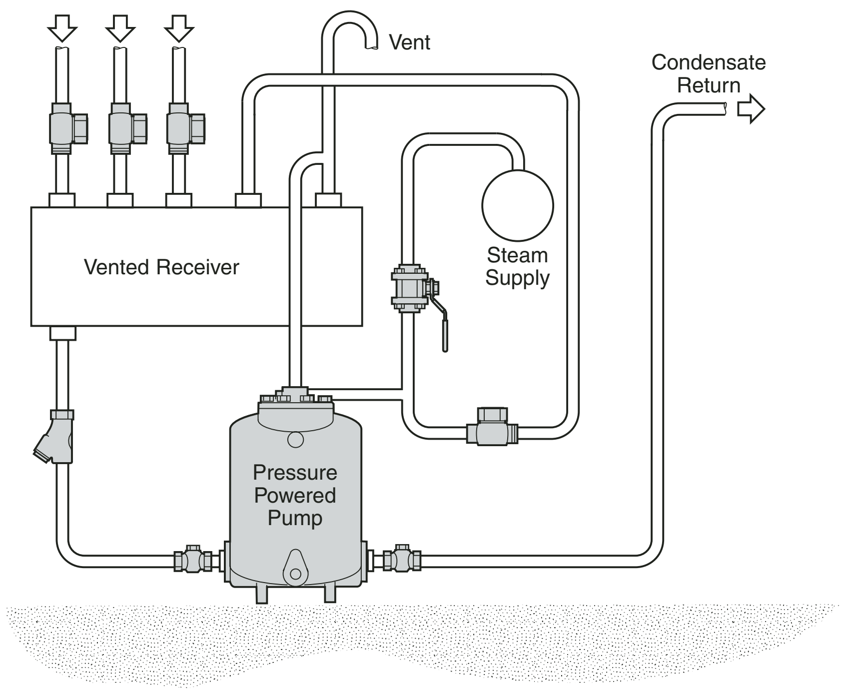


Figure 54
Venting of Pump Exhaust and Inlet Receiver Pipe in a Low Pressure System

Case in Action: Printing Mill Dryer Roll Drainage

Printing mills frequently mix toluene and isopropyl acetate with dyes to produce "quick drying" inks. This flammable mixture requires special care to avoid explosions and fire. Larger printing mills typically use steam-heated rolls to dry the printed material. The electric motor-driven condensate pumps that are commonly used, require explosion-proof controls/enclosures to accommodate the flammable atmosphere.

During a new project design, the consulting Engineer and Client decided to find a better way to deal with the hazardous environment and costly explosion-proof condensate pumps. The cost was of particular concern considering that the project included 16 dryer rolls on 2 printing machines, requiring 4 condensate pumps.

Solution

Four non-electric Pressure Powered Pumps™ were selected as alternatives to costlier electric pump sets. These were in addition to the 16 float and thermostatic steam traps installed on each dryer roll.

Benefits

- Installation cost was lower for the Pressure Powered Pumps™—no electrical wiring/controls required.
- Pressure Powered Pumps™ purchase price was substantially lower.
- Pressure Powered Pumps™ operation is safer than with electric pump/controls.
- Without mechanical seals, the Pressure Powered Pumps™ will operate with lower maintenance cost.

Clean Steam

The term "Clean Steam" can cover a wide range of steam qualities, depending on the production method used and the quality of the raw water.

The term "Clean Steam" is something of a misnomer and is commonly used as a blanket description to cover the three basic types - filtered steam, clean steam and pure steam.

a) Filtered steam is produced by filtering plant steam using a high efficiency filter. A typical specification would call for the removal of all particles greater than 2.8 microns, including solids and liquid droplets (Fig. 55).

b) Clean steam is raised in a steam generator or taken from an outlet on a multi-effect still, and is often produced from deionized or distilled water. A simplified generator and distribution system is shown in Fig. 56.

c) Pure steam is very similar to clean steam, but is always produced from distilled, deionized or pyrogen-free water, and is normally defined as "uncondensed water for injection (WFI)".

Often, the generic term "clean steam" is used to describe any of the three different types outlined above. It is therefore very important to know which is being used for any application, as the characteristics and system requirements for each can differ greatly. Note that in the following text, the expression "clean" steam will be used to denote any or all of the three basic types, where no differentiation is required.

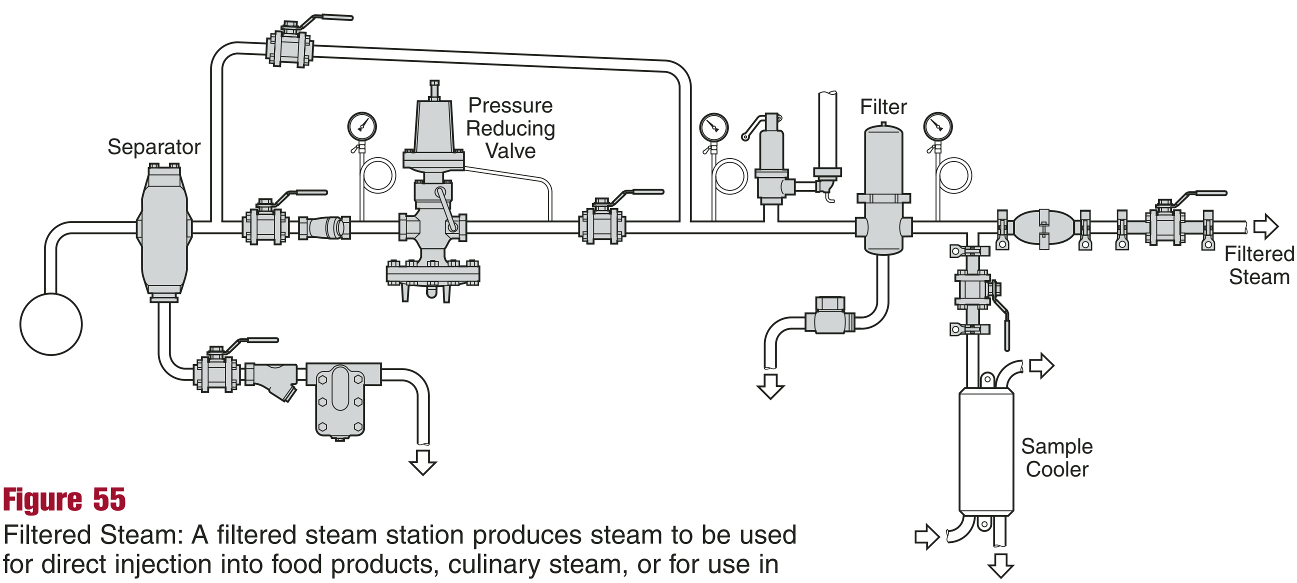


Figure 55

Filtered Steam: A filtered steam station produces steam to be used for direct injection into food products, culinary steam, or for use in sterilizers and autoclaves.

Steam Quality vs. Steam Purity

It is important to define the difference between steam quality and purity.

Steam Quality— "The ratio of the weight of dry steam to the weight of dry saturated steam and entrained water. For example, if the quality of the steam has been determined to be 95%, the wet-steam mixture delivered from the boiler is composed of 5 parts by weight of water, usually in the form of a fine mist, and 95 parts by weight of dry saturated steam. Likewise, if the quality of the steam has been determined to be 100%, there is no wet steam delivered from the boiler, 100% of the steam delivered from the boiler is dry saturated steam."

Steam Purity— "A quantitative measure of contamination of steam caused by dissolved solids, volatiles, or particles in vapour or by tiny droplets that may remain in the steam following primary separation in the boiler".

Table 14: Differences in Steam Characteristics

	Quality	Purity	
		Particles	Boiler Additives
Filtered	High	Typically 2.8 microns	Normally present
Clean	Varies on System Design	Varies	Limited to process
Pure	Varies on System Design	Varies	None

Thus, the three different types of "clean" steam (filtered, clean and pure) can, and will, have different characteristics, summarized in Table 14.

Note in particular that:

1. The quality of filtered steam will normally be high because water droplets larger than the filter element rating will be removed. Clean and pure steam systems will have a quality related to the design and operating characteristics of the generator, length and installation details of distribution system, insulation of system, number and effectiveness of mains, drainage points, etc.
2. Boiler additives may well be present in filtered steam and also possibly in clean steam, but often this will be limited by process requirements. For example, the FDA restricts the use of certain additives, including amines, in any steam which comes into direct contact with foods or dairy products.
3. Assuming the generating and distribution system have been designed and installed correctly, the particles present in a pure steam system will be water only. Dependent on feed water type, the same may also apply to clean steam systems.

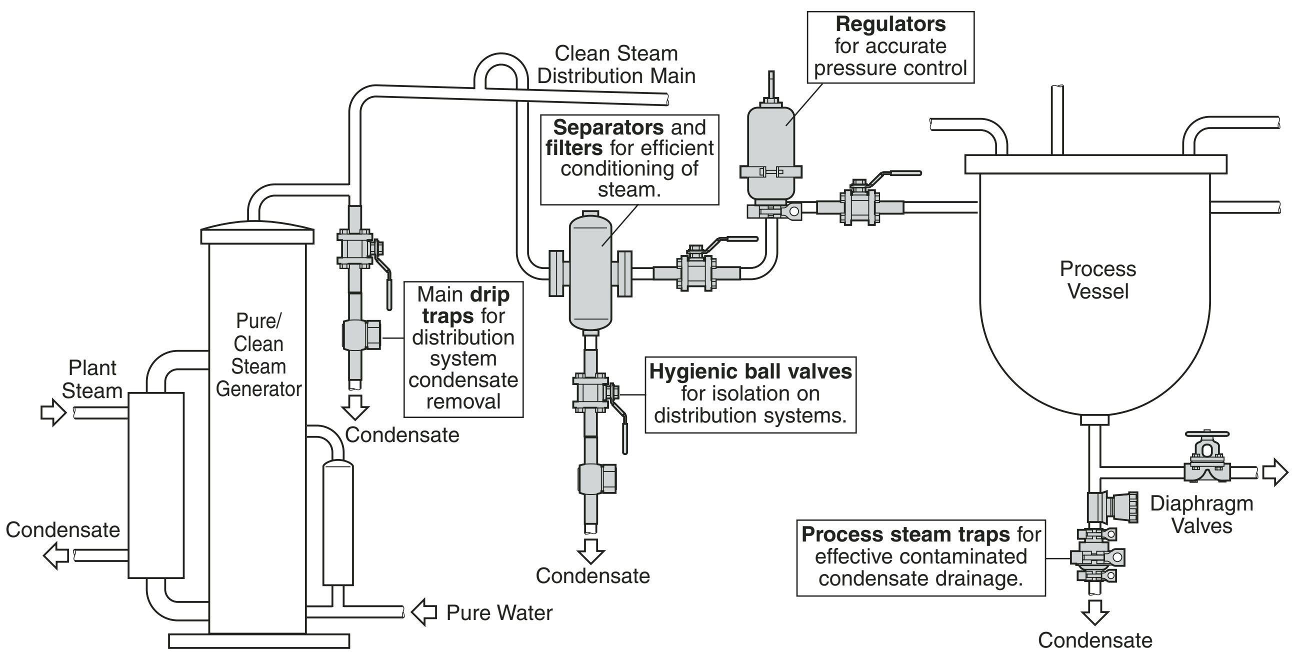


Figure 56

Clean/Pure Steam Generator and Distribution System

Overall Requirements of a "Clean" Steam System

The overall requirements of a "clean" steam system, irrespective of the means of generation of production used, can be very simply stated:

It is essential that the steam delivered to the point of use is of the correct quality and purity for the process.

In order to achieve this end goal, there are three key areas of design which must be considered once the requirement for clean steam has been identified.

- Point of Use
- Distribution
- Production

Design and operation of equipment, piping, components, etc. in all these three areas will influence the quality of the final process or products. It is essential for the needs of the user process to be the first concern. Must the steam be pyrogen free? Are any boiler additives allowed? Are products of corrosion going to harm the process or product? Must the risk of biological contamination be totally prevented? It is by answering these questions, and perhaps others, which will indicate the required type of production, design of the distribution system, and the operation modes of the user equipment, including aspects such as steam trapping.

Specific Requirements of "Clean" Steam Systems

Clean or pure steam produced from water of very high purity is highly corrosive or "ion hungry". The corrosive nature becomes more pronounced as the concentration of dissolved ions decreases with the resistivity approaching the theoretical maximum of 18.25 megohm/cm at 25°C. In order to recover a more natural ionic balance, it will attack many of the materials commonly used in pipework systems. To combat this, pipework, fittings, valves and associated equipment such as traps, must be constructed from corrosion resistant materials. Typically, a "clean" steam system of this type will have resistivity values of the condensate in the 2-15 megohm/cm range, resulting in very rapid attack of inferior quality components.

Even in some filtered plant steam applications, such as in the food, dairy and pharmaceuticals industries, certain corrosion inhibiting chemicals may be prohibited from the boiler and steam generating system. Again, condensate is then likely to be very aggressive and so careful consideration must be given to material selection.

A common problem encountered on clean and pure steam systems in the pharmaceutical industry is that of "rouging", which is a fine rusting of pipes and system components. This is encountered most frequently when low grade stainless steels are used, and further corrosion due to galvanic effects can take place where dissimilar alloys are present in the same system. Unless care is taken with material selection **throughout** the system, corrosion can become a major problem in terms of:

a) Contaminating the system with products of corrosion, which are undesirable or even potentially dangerous to the process or product.

b) Severely reduce life of system components, increasing maintenance time, material replacement costs, and system downtime.

In order to prevent these problems, austenitic stainless steel should be used throughout, never of lower grade than AISI 304. For severe duties, the recommended material is AISI 316 or 3161L (alternatively 316Ti) or better, passivated to further enhance corrosion resistance.

In summary, 316 or 316L stainless steel is **essential** in pure steam systems from its production at the generator right through to the steam traps. Not only will inferior materials corrode and fail prematurely, they will also lead to contamination of the system as a whole. Note that although filtered plant steam will not necessarily be so aggressive by nature, the exclusion of many of the corrosion inhibiting feed chemicals for end product purity reasons will still demand the use of austenitic stainless steel, never of lower grade than 304/304L, but preferable 316/316L.

Clean Steam and Condensate System Design

The proper and effective drainage of condensate from any steam system is good engineering practice, as it reduces corrosion, erosion, and waterhammer, and increases heat transfer. This becomes even more important in "clean" steam system, where poor condensate drainage in the distribution system or at the user equipment can result in rapid corrosion and also, under certain conditions, the risk of biological contamination. The following points should be carefully considered:

- Pipework should have a fall in the direction of flow of **at least** 1.0 inch in 10 ft., and should be properly supported to prevent sagging.
- Adequate mains and service pipe steam trapping should be provided, for example at all vertical risers, upstream of control valves, and at convenient points along any extended pipe length. Trapped drain points should be provided at intervals of at least every 100 ft.
- Undrained collecting points should not be used, as dirt should not be present and they provide an ideal location for bacterial growth where systems are shut down.
- Condensate should be allowed to discharge freely from steam traps using gravity and an air break. This air break should be provided at the manifold outlet or the closest convenient location (Fig. 57). Where the air break would otherwise be in a clean room, the potentially harmful

effects of flash steam can be prevented by using an expansion pot at the end of the manifold and venting through a filtered vent outside the clean room. The vent filter could alternatively be located at a kill tank, if used (Fig. 58).

- To prevent the risk of contamination, the direct connection of "clean" steam and condensate services should be prevented wherever possible. Under no circumstances should the condensate line or manifold lift above the level of the traps.
- Where the risk of biological contamination must be minimized, then care should be taken to select pipeline products which are self draining. This becomes most important in applications where the steam supply is frequently turned off, and where steam pipeline products are close coupled to sanitary process lines. Under these conditions, microbial growth will become possible in any pocket of condensate or process fluid retained in the system. However, where the steam supply is guaranteed, then this requirement does not become so stringent.
- Never "group trap" i.e. always use a single trap for draining each process line, vessel, etc. Failure to do this will invariably cause back-up of condensate in the system.
- The presence of crevices on pipe and component walls can provide an ideal location for microbial growth. Pipeline components which are likely to become fouled, such as steam traps installed on process systems, should be installed so they can be easily taken out of service for thorough cleaning.
- A "clean" steam service should **not** be interconnected to any other service which is not of sanitary design.
- Condensate from clean or pure steam systems should **not** be reused as make up for the clean/pure steam generation plant.
- Dead legs of piping which are not open to steam under normal operating conditions should be avoided by proper initial system design and the careful placement of isolation valves. Any dead leg open to steam must be properly trapped to prevent condensate build up.
- The use of OD tubing is becoming increasingly common for the distribution of clean steam. Table 15 gives capacities in lb/h for dry-saturated steam at various pressures. In order to reduce erosion and noise, it is recommended that designs should be based on flow velocities of 100 ft/sec. or less.

Typical Application

Sterile barriers, or block and bleed systems are used extensively in the biotechnology, pharmaceutical, food, dairy and beverage industries to prevent contaminating organisms from entering the process. A simple example also illustrating steam in place, condensate drainage from a process vessel, is shown in Fig. 59.

In this application, the steam trap is directly coupled to the process pipework, which is normally of sanitary design. It is quite possible that contamination at the trap, caused by either biological or chemical (corrosion) means, could find its way into the process system, thus resulting in failure of a product batch. Steam traps with corrosion resistant materials of construction and self draining features will reduce this risk, taking sanitary standards one step further from the process.

Due to piping arrangements, process fluids will often be flushed through the trap. This can often result in plugging if standard industrial designs of trap are used.

Specialty steam traps are called for which have the features outlined above plus the ability for rapid removal from the pipeline and quick disassembly for cleaning.

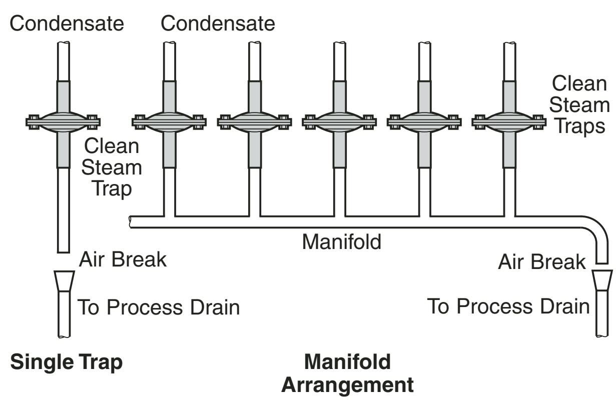


Figure 57
Steam Trap Discharge Details

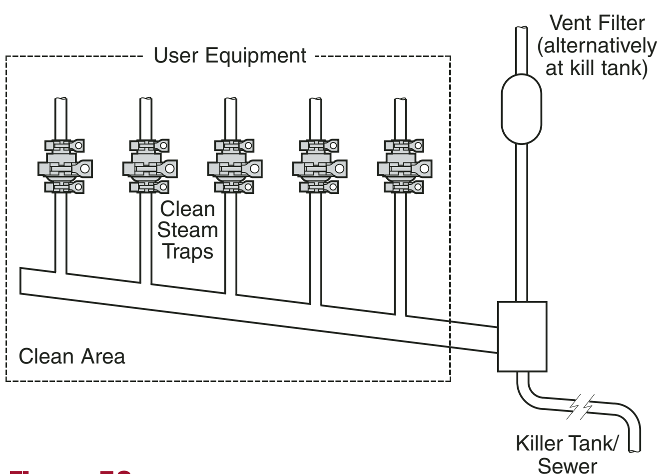


Figure 58
Steam Trap Discharge—Clean Room

Figure 59
Effective Condensate Drainage

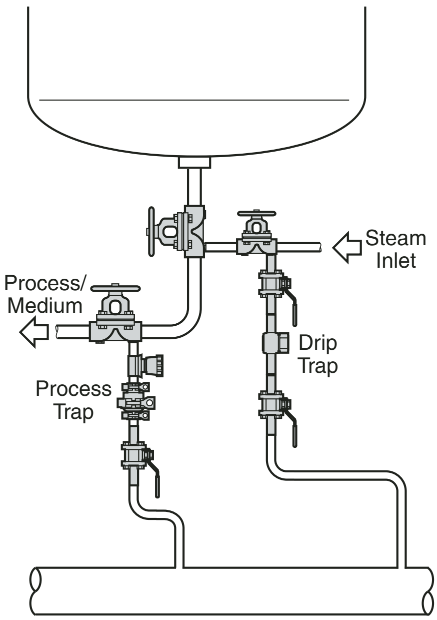


Table 15: Saturated Steam Capacities – OD Tube Capacities in lb/h

Pressure psi	Velocity ft/sec	Tube Size (O.D. x 0.065 inch wall) 1/4"	3/8"	1/2"	3/4"	1	1 1/2"	2"	2 1/2"	3"
5	50	—	—	5	20	35	90	170	270	395
	80	—	5	10	30	60	145	270	430	635
	120	—	5	15	45	85	215	405	650	950
10	50	—	5	10	25	45	110	210	335	490
	80	—	5	15	35	70	180	330	535	785
	120	—	10	20	55	110	270	500	800	1175
20	50	—	5	10	30	60	155	285	460	675
	80	—	10	20	50	100	245	460	735	1080
	120	5	10	25	75	150	370	685	1105	1620
30	50	—	5	15	40	80	195	365	585	855
	80	5	10	30	65	125	310	580	935	1370
	120	5	15	35	95	190	465	870	1400	2050
40	50	—	10	15	50	95	235	440	705	1035
	80	5	10	25	75	150	375	700	1125	1655
	120	5	20	40	115	230	556	1050	1690	2480
50	50	—	10	20	55	110	275	515	825	1210
	80	5	15	30	90	180	440	820	1320	1935
	120	5	20	50	135	265	660	1235	1980	2905
60	50	—	10	25	65	125	315	590	945	1385
	80	5	15	35	105	205	505	940	1510	2215
	120	5	25	55	155	305	755	1411	2265	3325
80	50	5	15	30	80	160	395	735	1180	1730
	80	5	20	45	130	255	630	1175	1890	2770
	120	5	30	70	195	380	950	1764	2835	4155
100	50	5	15	35	95	190	470	880	1415	2075
	80	5	25	55	155	305	755	1410	2265	3320
	120	10	35	85	230	455	1135	2115	3395	4975
120	50	5	20	40	115	220	550	1030	1650	2420
	80	5	30	65	180	355	885	1645	2640	3875
	120	10	40	95	270	535	1325	2465	3965	5810

Case in Action: Hospital Sterilizer

A hospital was experiencing continuing problems with a number of its sterilizers which were having an adverse effect on both the sterilization process itself and the amount of maintenance required to keep the units in service. These problems included:

- Ineffective sterilization
- Prolonged sterilization cycles
- Wet and discolored packs
- Instrument stains, spotting and rusting
- High maintenance of drain traps and controls
- Dirty sterilizer chambers requiring frequent cleaning

Solution

In discussion with the hospital maintenance engineer, the Spirax Sarco Sales Representative offered the opinion that these problems were a result of a wet and contaminated steam supply. In a number of cases the steam supply to the sterilizers was unconditioned, allowing moisture and solid particles, such as pipe scale and rust, to enter both the sterilizer jacket and chamber, resulting in the problems identified by the user.

The solution was to install Spirax Sarco steam filter stations. Each steam filter station is comprised of:

- An isolation valve to aid in maintenance.
- A separator complete with strainer and drain trap to remove residual condensate and any entrained moisture being carried in suspension within the steam.
- A main line strainer to remove larger solid particles.
- A steam filter and drain trap combination.

The steam filter specified was a Spirax Sarco CSF16 fitted with a 1 micron absolute filter element. The cleanable CSF16 filter element ensured that 99% of all particles larger than 0.1 microns were removed, while the thermostatic steam trap drained any condensate that formed in the filter body during operation and periods when the steam supply to the sterilizer was isolated. These installations resulted in steam supplies free of both moisture and solid particles.

Benefits

- Effective sterilization every time
- Reduced sterilization cycles and improved productivity
- High quality of packs and instruments without spots, stains or corrosion
- Minimal re-work of sterilizer loads
- Reduced cleaning and maintenance of sterilizer, drain traps and controls

Cost Savings

With the help of the hospital maintenance engineer, a steam filter station payback analysis sheet was completed. The estimated cost of maintenance, the cost associated with re-working wet or spotted packs, and the cost due to loss of performance were included in the payback calculation. In total, annual costs were over $25,000 for each sterilizer. Using this figure a payback period of less than two months was established for suitably sized Spirax Sarco steam filter stations.

Increasing attention is being paid in modern plants to means of assessing steam trap performance. While it is important to know if a trap is working normally or is leaking steam into the condensate return system, most of the available methods of assessing trap operation are of much more restricted usefulness than is appreciated. To explain this, it is necessary to consider the mode of operation of each type of trap when operating and when failed, and then to see if the proposed test method can distinguish between the two conditions.

Temperature Test Methods

One well established "method" of checking traps is to measure temperature, either upstream or downstream. People use pyrometers, remote scanners and temperature sensitive crayons or tapes, while generations of maintenance men have thought they could assess trap performance by spitting onto the trap and watching how the spittle reacted! Certainly, if a trap has failed closed, the temperature at the trap will be lower than normal, but equally the equipment being drained will also cool down. The trap is not leaking steam since it is closed, and this failure is only a cause of problems in applications like steam main drips where the condensate not discharged at the faulty trap is carried along the steam line. More usually, the temperature on the inlet side of the trap will be at or close to the saturation temperature of steam at whatever pressure is reaching the trap. Even if the trap were blowing steam, the temperature remains much the same.

The one exception is in the case of a temperature sensitive trap, especially one of the bimetal pattern. If this fails open, then the temperature at the inlet side will rise from the normal subcooled level to saturation values, and this rise may be detectable if the steam pressure is a known, constant value.

Measuring temperatures on the downstream side of a trap, by whatever method, is even less likely to be useful. Let's look first at a trap discharging through an open-ended pipe to atmosphere. The pressure at the trap outlet must be only just above atmospheric, and the temperature just above 212°F.

With any condensate present with the steam at temperatures above 212°F on the inlet side, the condensate, after passing through the trap will flash down to 212°F and this temperature is the one that will be found. Any leaking steam will help evaporate a little more of the condensate without increasing the temperature. Again, the only exception which may be encountered is the low pressure steam heating system where thermostatic traps normally discharge at temperatures below 212°F into atmospheric return. A temperature of 212°F here may indicate a leaking trap.

Discharge of condensate into a common return line is more usual than discharge to an open end, of course. The temperature in the return line should be the saturation temperature corresponding to the return pressure. Any increase in this temperature which may be detected will show that the return line pressure has increased. However, if trap "A" discharging into a line blows steam and the pressure in the line increases, then the pressure and temperature at traps "B" and "C" and all others on the line will also increase. Location of the faulty trap is still not achieved.

Visual Determinations

The release of flashing steam from condensate nullifies the effectiveness of test cocks, or three-way valves diverting a trap discharge to an open end for test purposes. It also restricts the information which can be gained from sight glasses. Consider a trap discharging to an open end some 500 lbs. per hour of condensate from a pressure of 125 psi. The steam tables show that each pound of water carries 324.7 BTU which is a144.5 BTU more than it can carry as liquid at atmospheric pressure. As the latent heat at 0 psig is 970.6 BTU/hr., then 144.5/970.6 lbs. of flash steam are released per pound of condensate, or 14.29%, which is some 74.45 pounds per hour. The volume of steam at 0 psig is 26.8 cu. ft. per pound, so some 1,995 cu. ft. per hour of flash steam is released. The remaining water, 500 - 74.45 = 425.55 lbs. has a volume of about 7.11 cu. ft. per hour. Thus, the discharge from the trap becomes 1995/1995 + 7.11 = 99.65% steam and 0.35% water, by volume.

It is sometimes claimed that an observer can distinguish between this "flash" steam and leakage steam by the color of the steam at the discharge point. While this may be possible when a trap is leaking steam but has no condensate load at all, so that only steam is seen at the discharge, it is obvious that the presence of any condensate will make such differentiation virtually impossible. It would be like trying to distinguish between 99.65% steam with 0.35% water, and perhaps 99.8% steam with 0.20% water!

Trap Discharge Sounds

In a closed piping system, trap discharge sounds may be a good indicator of its operation. A simple stethoscope will be of little value, but the sound produced at ultra-high frequencies measured by an ultrasonic instrument eliminates background noise interference. Live steam flow produces a greater and steady level of ultrasound, while flashing condensate tends to have a crackling sound and the level changes with the trap load. The problem is that the instrument requires the operator to make a judgement as to trap condition which will only be as reliable as his training and experience provide for.

Table 16: Steam Trap Discharge Modes

	Mode of Operation				
Trap Type	No Load	Light Load	Normal Load	Full or Overload	Usual Failure Mode
Float & Thermosatic	No Action	Usually continuous but may cycle at high pressure		Continuous	Closed, A.V. Open
Inverted Bucket	Small Dribble	Intermittent	Intermittent	Continuous	Open
Balanced Pressure Thermostatic	No Action	May Dribble	Intermittent	Continuous	Variable
Bimetallic Thermostatic	No Action	Usually Dribble Action	May blast at high pressures	Continuous	Open
Impulse	Small Dribble	Usually continuous with blast at high loads		Continuous	Open
Disc Thermo-Dynamic	No Action	Intermittent	Intermittent	Continuous	Open

What must be done, using all audible and visual clues, is to detect normal or abnormal cycling of the discharge. Even this method is very fallible, since the mode of operation of different trap types if not nearly so well defined as is sometimes thought. Table 16 lists some of the possibilities and allows the problem to be seen more clearly.

It is seen that the "signal" to be obtained from the trap, whether visual, audio or temperature, is usually going to be so ambiguous as to rely largely on optimism for interpretation. The one trap which is fairly positive in its action is the disc thermodynamic type—if this is heard or seen to cycle up to ten times per minute, it is operating normally. The cycling rate increases when the trap becomes worn and the characteristic "machine gun" sound clearly indicates the need for remedial action.

Spira-tec Leak Detector System

Logic says that if it is not possible to have a universally applicable method of checking steam traps by examining the traps themselves, then we must see if it can be done by checking elsewhere. This is what Spirax Sarco has done with the Spira-tec system. See Fig. 61 (page 58).

The Spira-tec detector chamber is fitted into the condensate line on the inlet side of the trap. If there is, at this point, a normal flow of condensate towards the trap, together with a small amount of air and the steam needed to make up heat loss from the body of the steam trap, then all is normal. On the other hand, an increased flow of gas along the pipe indicates that the trap is leaking.

The chamber contains an inverted weir. Condensate flows under this weir and a small hole at the top equalizes the pressure on each side when the steam trap is working normally. An electrode on the upstream side of the baffle detects the presence of condensate by its conductivity which is much higher than that of steam. By plugging in the portable indicator, it is possible to check if the electrical circuit is complete when a visual signal indicates that the trap is working.

If the trap begins to leak steam, then the pressure on the downstream side of the weir begins to fall. The higher pressure on the upstream side drops the condensate level below the electrode and exposes it to steam. The "conductivity" circuit is broken and the indicator light gives a "fail" signal.

The advantage of the system lies in the very positive signal which does not require experience of personal judgement before it can be interpreted.

Using suitable wiring, the test point can be located remote from the sensor chamber or it can have a multi switch to allow up to twelve (12) chambers to be checked from a single test location. When appropriate, an electronic continuous 16-way checking instrument can monitor the chambers and this is readily connected into a central Energy Management System.

The object of detecting leaking steam traps is to correct the problem. This can mean replacement of the whole trap, or perhaps of the faulty part of the internal mechanism. It is very useful indeed to be able to check a repaired trap in the workshop before it is installed in the line, and many repair shops now use a Spira-tec chamber as part of a bench test rig. The diagram shows a simple hookup which allows suspect or repaired traps to be positively checked. (Fig. 60)

Cost Of Steam Leaks

The installation and use of the Spira-tec units does involve some cost, and it is necessary to compare this with the cost of steam leakages to see if the expenditure is economically justifiable. Since all equipment must wear and eventually fail, we need first an estimate of the average life of a steam trap. Let us assume that in a particular installation, this is,

say seven (7) years. This means that after the first seven years of the life of the plant, in any year an average of almost 15% of the traps will fail. With an annual maintenance campaign, some of the traps will fail just after being checked and some just before the next check. On average, the 15% can be said to have failed for half the year, or 7-1/2% of traps failed for the whole year.

Now, most of the traps in any installation, on the mains drip and tracer installations are probably 1/2" or 3/4" size and most of them are oversized, perhaps by a factor of up to 10 or more. Let us assume that the condenste load is as high as 25% of the capacity of the trap. If the trap were to fail wide open, then some 75% of the valve orifice would be available for steam flow. The steam loss then averages 75% of 7-1/2% of the steam flow capacity of the whole trap population, or about 5.62%.

The steam flow through a wide open seat clearly depends on both pressure differentials and orifice sizes, and orifice sizes in a given size of trap such as 1/2" usually are reduced as the designed working pressure increases.

Estimating Trap Steam Loss

Steam loss through a failed open trap blowing to atmosphere can be determined from a variant of the Napier formula as follows:

Steam Flow in lbs/hr =
$$24.24 \times Pa \times D^2$$

Where:

Pa = Pressure in psi absolute

D = Diameter of trap orifice in inches

By multiplying the steam loss by hours of operation, steam cost (typically $6.00 per 1,000 pounds), and by the number of failed traps, total cost of steam system loss may be estimated.

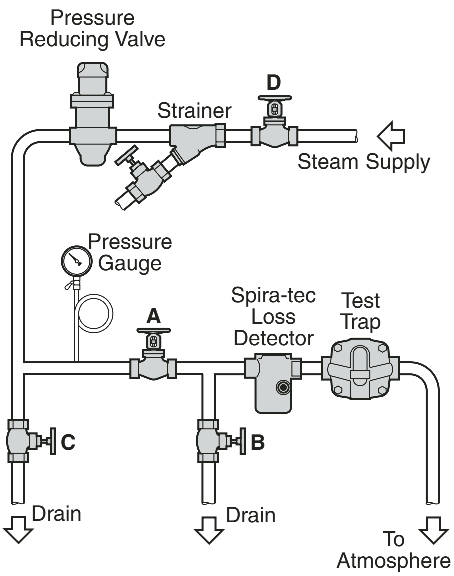
The formula above should not be used to directly compare potential steam loss of one type of trap against another because of differences in failure modes. In those that fail open only the inverted bucket trap orifice blows full open. Thermostatic types usually fail with their orifice at least partially obstructed by the valve, and flow through thermodynamic types is a function of many passageways and must be related to an equivalent pass area. In every case, no trap begins losing steam through wear or malfunction until the leakage area exceeds that needed by the condensate load. The cost then begins and reaches the maximum calculated only when the trap fails completely. The object is, of course to prevent it from reaching that stage. The steam system always functions best when traps are selected that are best for the application and checked on a regular basis to control losses.

Table 17: Steam Flow through Orifices Discharging to Atmosphere

Diameter (inches)	Steam flow, lb/h, when steam gauge pressure is: 2 psi	5 psi	10 psi	15 psi	25 psi	50 psi	75 psi	100 psi	125 psi	150 psi	200 psi	250 psi	300 psi
1/32	.31	.47	.58	.70	.94	1.53	2.12	2.7	3.3	3.9	5.1	6.3	7.4
1/16	1.25	1.86	2.3	2.8	3.8	6.1	8.5	10.8	13.2	15.6	20.3	25.1	29.8
3/32	2.81	4.20	5.3	6.3	8.45	13.8	19.1	24.4	29.7	35.1	45.7	56.4	67.0
1/8	4.5	7.5	9.4	11.2	15.0	24.5	34.0	43.4	52.9	62.4	81.3	100	119
5/32	7.8	11.7	14.6	17.6	23.5	38.3	53.1	67.9	82.7	97.4	127	156	186
3/16	11.2	16.7	21.0	25.3	33.8	55.1	76.4	97.7	119	140	183	226	268
7/32	15.3	22.9	28.7	34.4	46.0	75.0	104	133	162	191	249	307	365
1/4	20.0	29.8	37.4	45.0	60.1	98.0	136	173	212	250	325	401	477
9/32	25.2	37.8	47.4	56.9	76.1	124	172	220	268	316	412	507	603
5/16	31.2	46.6	58.5	70.3	94.0	153	212	272	331	390	508	627	745
11/32	37.7	56.4	70.7	85.1	114	185	257	329	400	472	615	758	901
3/8	44.9	67.1	84.2	101	135	221	306	391	476	561	732	902	1073
13/32	52.7	78.8	98.8	119	159	259	359	459	559	659	859	1059	1259
7/16	61.1	91.4	115	138	184	300	416	532	648	764	996	1228	1460
15/32	70.2	105	131	158	211	344	478	611	744	877	1144	1410	1676
1/2	79.8	119	150	180	241	392	544	695	847	998	1301	1604	1907

Figure 60

Steam Trap Test Rig



Inexpensive test stand may be used to test steam trap operation. Valves A, B, C, and D are closed and the trap is attached. Valve C is cracked and valve D is slowly opened. The pressure-reducing valve is adjusted to the rated pressure of the trap being tested, valve C is closed, and valve A is opened slowly, allowing condensate flow to the trap until it is discharged. Valve B is then partially opened to allow the condensate to drain out, unloading the trap. Under this final condition, the trap must close with a tight shutoff. With some trap configurations, a small amount of condensate may remain downstream of the trap orifice. Slow evaporation of this condensate will cause small amounts of flash steam to flow from the discharge of the trap even though shutoff is absolute.

Spira-tec Trap Leak Detector System for Checking Steam Traps

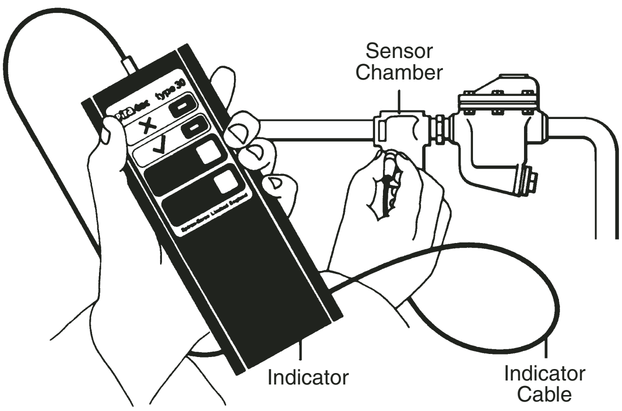
Purpose

The Spira-tec Trap Leak Detector System is designed to indicate if a steam trap is leaking steam. It can be used to check any known type or make of trap while it is working.

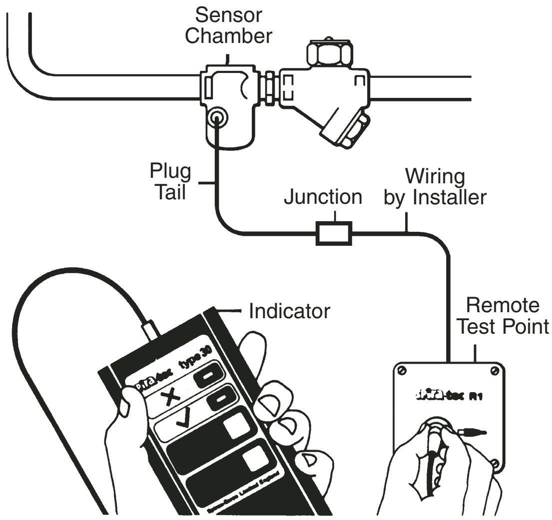
Equipment

1. Sensor chamber fitted immediately upstream of the trap (close coupled), the same size as the trap.
2. Indicator with cable.
3. Where the sensor chamber is not readily accessible, a Remote Test Point may be fitted at a convenient position, wired back through a junction to the sensor chamber. Remote Test Points for either one chamber or up to 12 chambers, are available.
4. An Automatic Remote Test Point, capable of interfacing with most Building Management Systems, is also available allowing up to 16 steam traps to be continuously scanned for steam wastage.

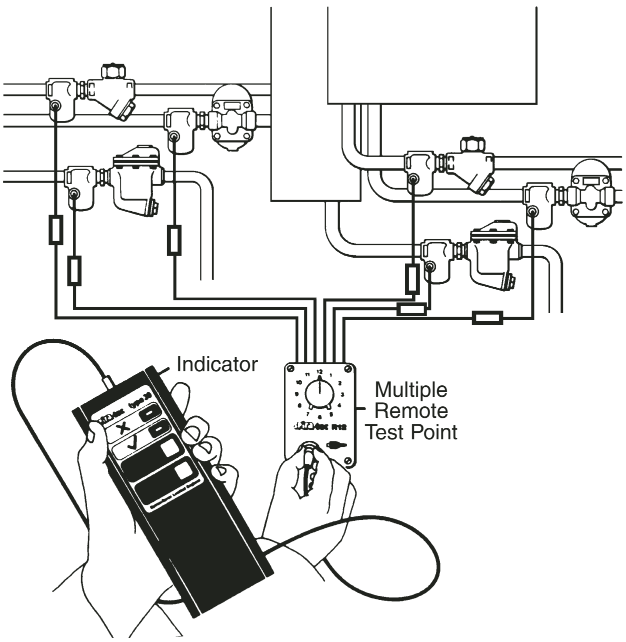
Figure 61



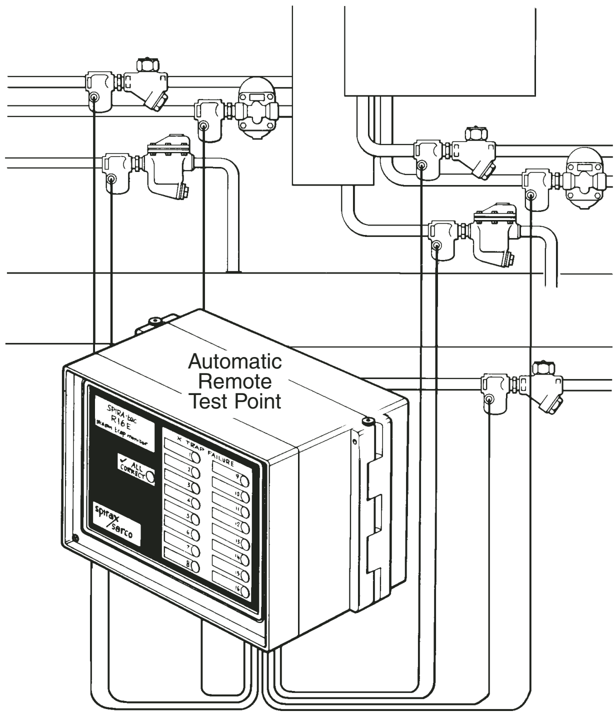
Basic steam trap checking system—
Spira-tec sensor chamber, indicator and indicator cable.



Single inaccessible steam trap checking system—Spira-tec sensor chamber, plug tail, wiring (by installer), remote test point, indicator and indicator cable.



Multiple inaccessible steam trap checking system— Spira-tec sensor chambers, plug tails, wiring (by installer), remote multiple test point, indicator and indicator cable.



Continuous scanning system—Spira-tec Automatic Remote test point, sensor chambers, plug tails.

The steam meter is the basic tool that provides an operator or manager, information vital in monitoring and maintaining high efficiency levels within a plant or building. This information can be split into four categories:

Plant Efficiency

- Is idle machinery switched off?
- Is the plant loaded to capacity?
- Is plant efficiency deteriorating over time indicating the need for cleaning, maintenance and replacement of worn parts?
- When do demand levels peak and who are the major users? This information may lead to a change in production methods to even out steam usage and ease the peak load problems on boiler plant.

Energy Efficiency

- Is an energy saving scheme proving effective?
- How does the usage and efficiency of one piece of plant compare with another?

Process Control

- Is the optimum amount of steam being supplied to a certain process?
- Is that steam at the correct pressure and temperature?

Costing and Custody Transfer

- How much steam is being supplied to each customer?
- How much steam is each department or building within an organization using?

Selecting a Steam Meter

Before selecting a steam meter it is important to understand how a meter's performance is described. The overall performance of a meter is a combination of Accuracy, Repeatability and Turndown.

Accuracy

This is the measurement (expressed as a percentage) of how close the meter's indication of flow is to the actual flow through the meter. There are two methods used to express accuracy (or percentage of uncertainty) and they have very different meanings.

a. Measured Value or Actual Reading

Example: Meter is ranged 0-1000 lb/h and has a specified accuracy of ± 3% of Actual Reading

At an indicated flow rate of 1,000 lb/h, the true flow rate lies between 1,030 and 970 lb/h.

At an indicated flow rate of 100 lb/h, the true flow rate lies between 103 and 97 lb/h.

b. F.S.D. or Full Scale Deflection

Example: Meter is ranged 0-1000 lb/h and has a specified accuracy of ± 3% FSD

At an indicated flow rate of 1,000 lb/h, the true flow rate lies between 1,030 and 970 lb/h.

At an indicated flow rate of 100 lb/h, the true flow rate lies between 130 and 70 lb/h (i.e. ± 30% of Reading !).

Repeatability

This describes the ability of a meter to indicate the same value for an identical flowrate over and over again. It should not be confused with accuracy i.e. the meter's repeatability may be excellent in that it shows the same value for an identical flowrate on several occasions, but the reading may be consistently wrong (or inaccurate). Repeatability is expressed as a percentage of either actual reading or FSD. Good repeatability is important for observing trends or for control e.g. batching.

Turndown

Sometimes called Turndown Ratio, Effective Range or even Rangeability. In simple terms, it is the range of flow rate over which the meter will work within the accuracy and repeatability tolerances given. If a meter works within a certain specified accuracy at a maximum flow of 1,000 lb/h and a minimum flow of 100 lb/h, then dividing the maximum by the minimum gives a turndown of 10:1. A wide turndown is particularly important when the flow being measured is over a wide range. This could be due to a variation in process e.g. a laundry could be operating 1 machine or 20 machines (20:1 turndown), or due to seasonal variations in ambient temperature if the steam is being used for space heating - the difference in demand between mid winter and mid summer can be considerable. Generally the bigger the turndown the better.

The Spirax Sarco family of meters covers a wide range of sizes, turndown, accuracy, and repeatability as detailed in Table 18.

Table 18: Specification of the Spirax Sarco Meter Range

Meter Type	Sizes	Accuracy	Turndown	Repeatability
Orifice Plate	1" -24"	± 3% of Reading	4:1	± 0.3% of Reading
Gilflo (Made to Order)	2" - 16"	±1% FSD (± 1% of Reading with flow computer)	100:1	± 0.25% of Reading
Gilflo (S.R.G.)	2" - 8"	± 2% FSD (± 1% of Reading with flow computer)	100:1	± 0.25% of Reading
Gilflo (I.L.V.A.)	2" - 8"	± 2% FSD (± 1% of Reading with flow computer)	100:1	± 0.25% of Reading
Spiraflo	1-1/2" - 4"	± 2% of Reading (50% - 100% of meter range) ± 1% FSD (1% - 50% of meter range)	25-40:1	± 0.5% of Reading

Density Compensation

For accurate metering of compressible fluids such as gases and vapors, the actual flowing density must be taken into account. This is especially true in the case of steam. If the actual flowing density of the steam is different to the specified density for which the meter was originally set up or calibrated, then errors will occur. These errors can be considerable and depend on both the magnitude of difference between the specified density and the actual flowing density, and the type of meter being used.

Example:

The steam meter is set up and calibrated for 100 psig (specific volume = 3.89 ft^3/lb)

The steam is actually running at a pressure of 85 psig (specific volume = 4.44 ft^3/lb).

a. Differential Pressure Device
(e.g. Orifice Plate or Gilflo Meter)

$$\text{Error} = \left[\sqrt{\frac{\text{s.v. actual}}{\text{s.v. specified}}} - 1\right] \times 100$$

$$\text{Error} = \left[\sqrt{\frac{4.44}{3.89}} - 1\right] \times 100$$

Error = 6.8 Therefore the meter will over read by 6.8%

b. Velocity Device
(e.g. Vortex Meter)

$$\text{Error} = \left[\frac{\text{s.v. actual}}{\text{s.v. specified}} - 1\right] \times 100$$

$$\text{Error} = \left[\frac{4.44}{3.89} - 1\right] \times 100$$

Error = 14 Therefore the meter will over read by 14%

In the case of saturated steam, pressure and temperature are related and therefore to establish the flowing density of saturated steam, either pressure or temperature should be measured. In the case of superheated steam, pressure and temperature can vary independently from one another and therefore to compensate for changes in density of superheated steam, both pressure and temperature must be measured.

Installation

Ninety percent of all metering failures or problems are installation related. Care should be taken to ensure that not only is the meter selected suitable for the application, but that the steam is correctly conditioned both to improve meter performance and provide a degree of protection, and that the manufacturer's recommendations regarding installation are carefully followed.

Steam Conditioning

For accurate metering of saturated steam, irrespective of the meter type or manufacturer, it is important to condition the steam so that it is in the form of, or as close as possible to, a dry gas. This can be achieved by correct steam engineering and adequate trapping to reduce the annular film of water that clings to the pipe wall, and effective separation ahead of the meter to remove much of the entrained droplets of water. It is therefore recommended that a steam conditioning station (as shown in Fig. 62) is positioned upstream of any type of meter on saturated steam applications. This will enhance accuracy and protect the meter from the effects of water droplets impacting at high velocity. Good steam engineering such as the use of eccentric reducers, effective insulation and adequate trapping will also prevent the dangerous effects of high velocity slugs of water known as water-hammer which can not only destroy meters but will also damage any valves or fittings in it's path.

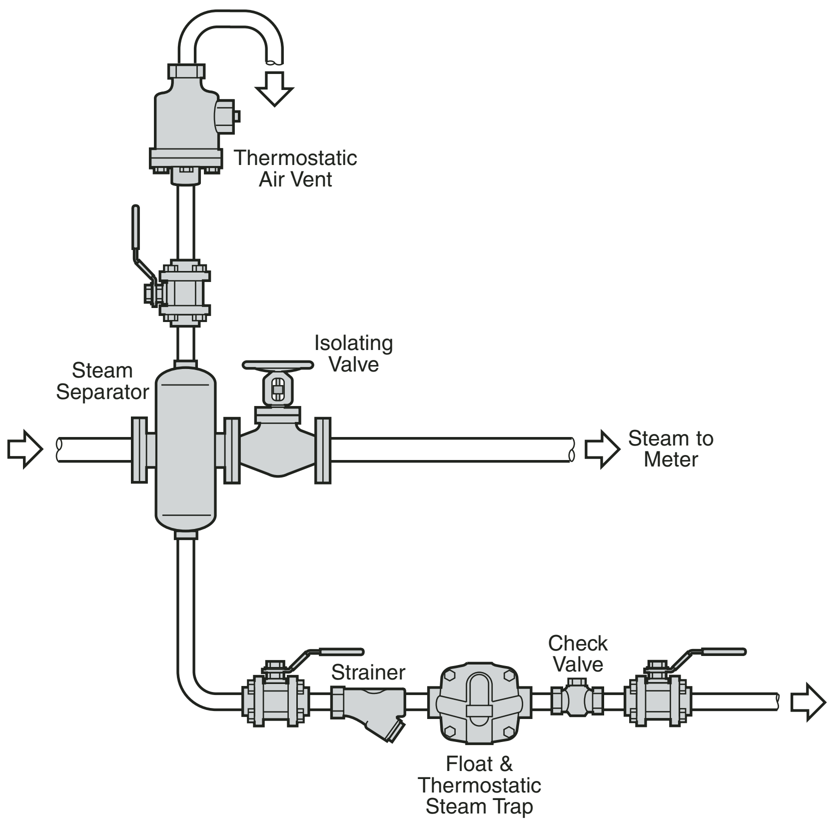


Figure 62
Steam Conditioning Station

Meter Location

Meters need to be installed in defined lengths of straight pipe to ensure accurate and repeatable performance. These pipe lengths are usually described in terms of the number of pipe diameters upstream and downstream of the meter. For example, an Orifice Plate with a Beta ratio of 0.7 installed after a 90° bend requires a minimum of 28 pipe diameters of straight pipe upstream and 7 downstream. If the pipe diameter is 6", this is equivalent to 14 feet upstream and 3-1/2 feet downstream.

If the meter is located downstream of two 90º bends in different planes, then the minimum straight length required upstream of the meter is 62 pipe diameters or thirty one feet. This can be difficult to achieve, particularly in fairly complex pipework systems, and there may not in fact be a location that allows these criteria to be met. This is an important consideration when selecting a meter.

Table 19 shows the minimum piping requirements for Orifice Plates as laid down in the US standard ASME MFC-3M together with the manufacturers recommendations for vortex and spring loaded variable area meters. See Figures II-93, 94, 95, 96 (pages 131 and 132).

Table 19: Recommended Minimum Straight Lengths (D) for Various Meter Types

On Upstream (inlet) side of the primary device									Downstream
Meter Type	ß Ratio [3]	Single 90° Bend	Two 90° Bends Same Plane	Two or more 90° Bends Different Planes	Reducer 2D to D	Expander 0.5D to D	Globe Valve Fully Open	Gate Valve Fully Open	All Fittings in this table
Orifice Plate	0.30	10	16	34	5	16	18	12	5
Orifice Plate	0.35	12	16	36	5	16	18	12	5
Orifice Plate	0.40	14	18	36	5	16	20	12	6
Orifice Plate	0.45	14	18	38	5	17	20	12	6
Orifice Plate	0.50	14	20	40	6	18	22	12	6
Orifice Plate	0.55	16	22	44	8	20	24	14	6
Orifice Plate	0.60	18	26	48	9	22	26	14	7
Orifice Plate	0.65	22	32	54	11	25	28	16	7
Orifice Plate	0.70 [4]	28	36	62	14	30	32	20	7
Orifice Plate	0.75	36	42	70	22	38	36	24	8
Orifice Plate	0.80	46	50	80	30	54	44	30	8
Vortex [1]	N/A	20 - 40	20 - 40	40	10 - 20	10 - 35	50	20 - 40	5 - 10
Spiraflo [2]	N/A	6	6	12	6	12	6	6	3 - 6
Gilflo [2]	N/A	6	6	12	6	12	6	6	3 - 6
Gilflo SRG [2]	N/A	6	6	12	6	12	6	6	3 - 6
Gilflo ILVA [2]	N/A	6	6	12	6	12	6	6	3 - 6

Notes:
1. The table shows the range of straight lengths recommended by various Vortex meter manufacturers.
2. Downstream requirements are 3D and 6D when upstream are 6D and 12D respectively.
3. ß ratio = Orifice diameter (d) divided by Pipe diameter (D)
4. Most Orifice Plates are supplied with a ß ratio of around 0.7 which gives the best pressure recovery without compromising signal strength.

Air Compressors

Heat is released when air or any gas is compressed. The compressor must be cooled to avoid overheating, usually by circuating water through the jackets. Cooling is an important function which must be controlled to ensure maximum efficiency. Overcooling wastes water and leads to condensation within the cylinders, with deterioration of the lubricating oils. Undercooling reduces compressor capacity and can result in serious damage to the compressor. Automatic temperature control of cooling water flow ensures maximum efficiency.

The atmosphere is a mixture of air and water vapor. Free air has a greater volume, and moisture holding capacity, than compressed air at the same temperature. As the compressed air is cooled after leaving the compressor, or between stages, some of the water is precipitated. This water must be drained from the system to avoid damage to pneumatic valves and tools.

Choice Of Drainer Trap

The quantities of water which must be drained from the air are relatively small, even on quite large installations, providing they are dealt with continuously. It is unusual to need air traps in sizes larger than 1/2". Except where a worn compressor is allowing lubricating oils to be discharged with the compressed air, float operated drainers are the best choice.

Where the presence in the system of water/oil emulsions interferes with the operation of float drainers, the thermodynamic TD trap is used. As the TD trap needs an operating pressure of at least 50 psi when used as an air drainer, care must be taken when it is used on small systems. Preferably, the TD's should be valved off at start-up until the system pressure is up to 50 psi or more.

Sizing Compressed Air Traps

The amount of water which is to be discharged is determined from steam table saturated vapor density or estimated with the help of a graph, Fig. 63 and compression ratio table. An example shows how this is used.

Example:

How much water will precipitate from 150 cfm of free air at 70°F and 90% relative humidity when compressed to 100 psig and cooled to 80°F?

Air flow = 150 cfm X 60 = 9000 cu. ft/hour.
From Fig. 63, at 70°F water in air drawn in will be

$$1.15 \times \frac{9000}{1000} \times 90\% = 9.32 \text{ lb/h}$$

Determine excess moisture due to compression by dividing hourly air flow by factor from Compession Ratio Table 20B (page 64), and convert for (absolute) temperature.

Compression ratio at 100 psig = 7.8

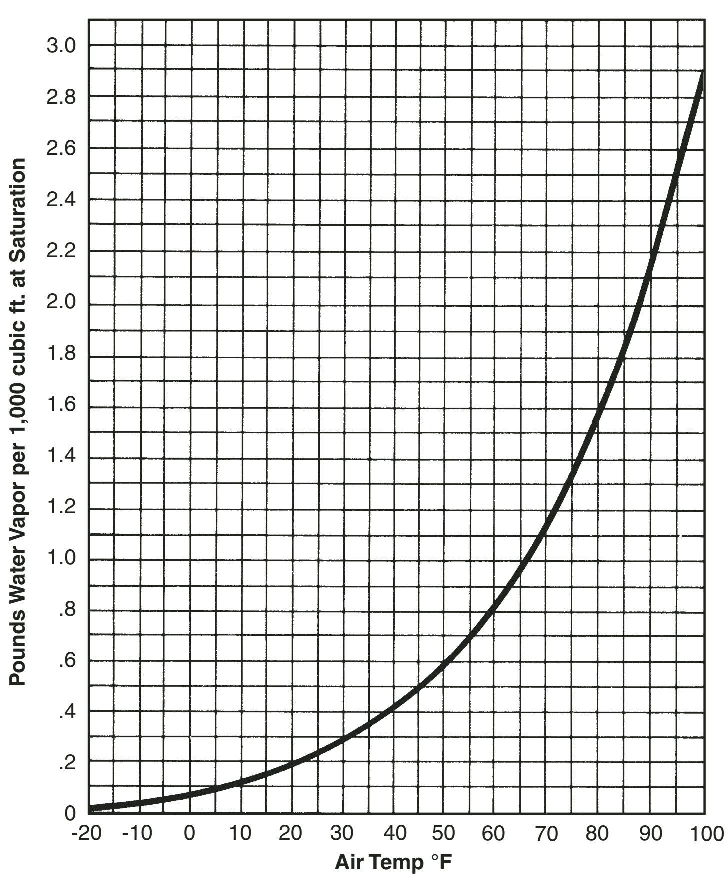
$$\text{Air volume after compression} = \frac{9000}{7.8} \times \frac{(460 + 80)}{(460 + 70)} = 1175 \text{ cu. ft./h}$$

From Fig. 63, 1000 cu. ft. at 80°F can carry 1.6 lb. of water.
1175 cu. ft. will carry

$$\frac{1175}{1000} \times 1.6 = 1.88 \text{ lb/h}$$

So, (9.32 lb. – 1.88 lb.) = 7.44 lb/h of water will separate out.

Figure 63: Moisture Holding Capacity of Air at Varying Temperatures



Drainer Installation

Automatic drainers are needed at any absorption or refrigerant dryer, and any separator which is instaled in the air line from the aftercooler, or at the entry to a building. They are also needed at the low points in the distribution lines. (Fig. 64)

Unless fitted close to the points being drained, and on light loads, drainers often need a balance line to allow air to be displaced from the piping or the drainer body as water runs in. The balance line is connected above the drain point, and should not be upstream of it. See Fig. II-115 (page 140).

Distribution Lines

These form the all important link between the compressor and the points of usage. If they are undersized, the desired air flow will be accompanied by a high pressure drop. This necessitates extra power input at the compressor. For example, a pressure at the compressor of 120 psi where a pressure of 100 psi would have sufficed without a high pressure drop in the lines, needs an additional power input of 10%.

The correct size of compressed air lines can be selected by using Fig. 66 on page 66.

Example: 1,000 cu. ft. of free air per minute is to be transmitted at 100 PSIG pressure through a 4" line standard weight pipe. What will be the pressure drop due to friction?

1. Enter the chart at the top at the point representing 100 psig pressure.
2. Proceed vertically downward to the intersection with horizontal line representing 1,000 CFM.
3. Next proceed parallel to the diagonal guide lines to the right (or left) to the intersection with the horizontal line representing a 4" line.
4. Proceed vertically downward to the pressure loss scale at the bottom of the chart. You will note that the pressure loss would be 0.225 psi per 100 ft. of pipe.

It is usual to size compressed air lines on velocity, while keeping a watchful eye on pressure drop.

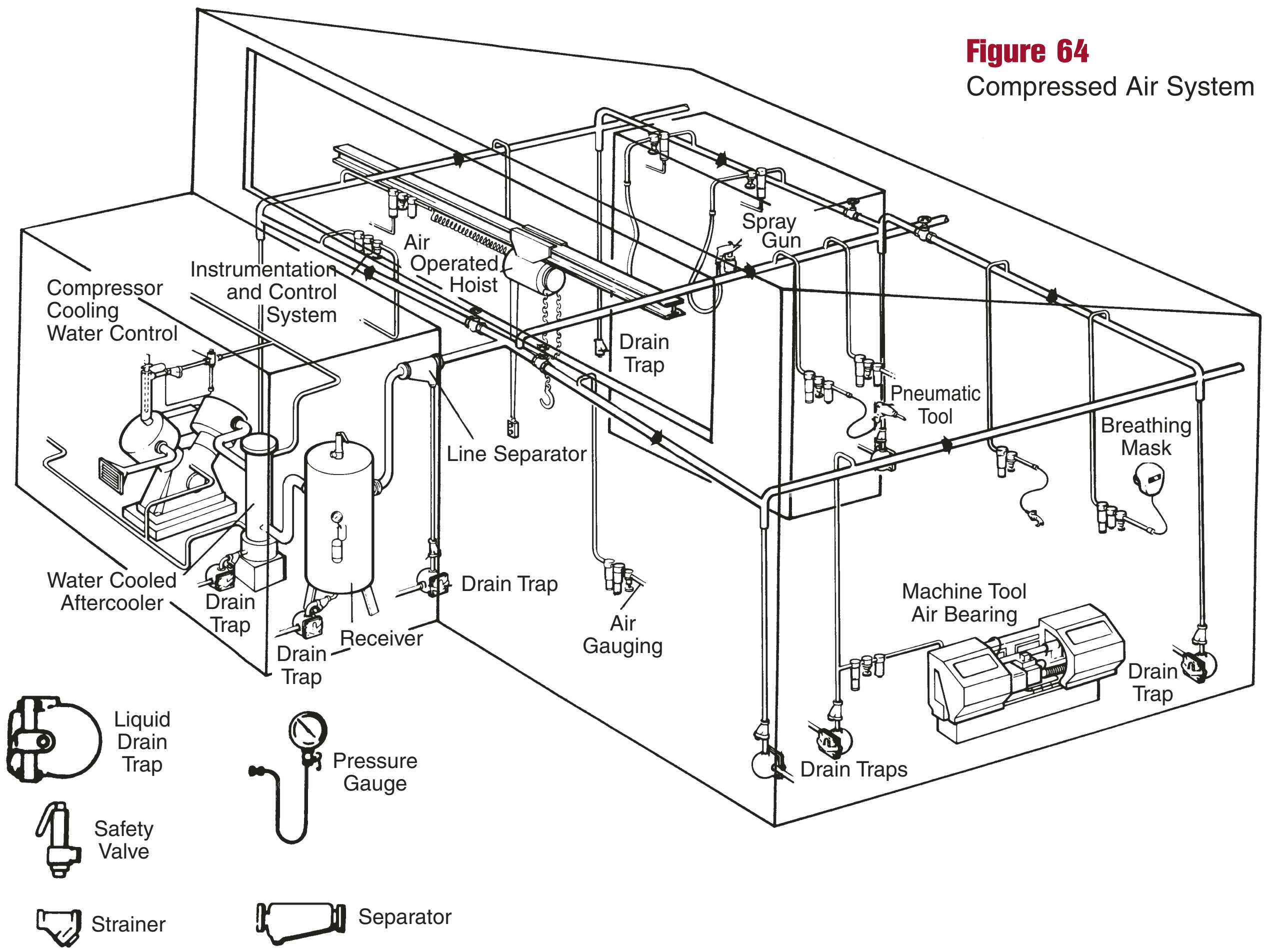


Figure 64
Compressed Air System

Table 20A: Pumped Circulation Water Storage Tanks

Compressor Capacity, cfm free air	25	50	100	150	200	300	450	600	800
Tank Capacity, gallons	50	100	180	270	440	550	850	1000	1200

Table 20B: Ratio of Compression

Gauge Pressure psi	10	20	30	40	50	60	70	80
Ratio of Compression	1•68	2•36	3•04	3•72	4•40	5•08	5•76	6•44
Gauge Pressure psi	90	100	110	120	130	140	150	200
Ratio of Compression	7•12	7•8	8•48	9•16	9•84	10•52	11•2	14•6

Table 21: Cooling Water Flow Rates

Compressor operating at 100 psi	Water Flow per 100 cfm free air
Single Stage	1.2 gpm
Single Stage with Aftercooler	4.8 gpm
Two Stage	2.4 gpm
Two Stage with Aftercooler	6 gpm

An air velocity of 20 to 30 ft/second or 1200 to 1800 ft/minute, is sufficiently low to avoid excessive pressure loss and to prevent re-entrainment of precipitated moisture. In short branches to the air-using equipment, volocities up to 60-80 ft/second or 3600-4800 ft/minute are often acceptable.

Checking Leakage Losses

Air line leaks both waste valuable air and also contribute to pressure loss in mains by adding useless load to compressors and mains. Hand operated drain valves are a common source of leakage that can be stopped by using reliable automatic drain traps. Here is a simple way of making a rough check of leakage loss. First, estimate the total volume of system from the receiver stop valve to the tools, including all branches, separators, etc. Then with no equipment in use, close the stop valve and with a stop watch note the time taken for the pressure in the system to drop by 15 psi. The leakage loss per minute is:

$$\text{Cu. ft. of Free Air Loss per Minute} = \frac{\text{Volume of System Cu. Ft.}}{\text{Time in Minutes to Drop Pressure 15 psi}}$$

Compressor Cooling

Air cooled compressors, formerly available only in the smaller sizes, are now found with capacities up to 750 cfm, and rated for pressures up to 200 psi. The cylinders are finned and extra cooling is provided by arranging the flywheel or a fan to direct a stream of air on to the cylinder. Such compressors should not be located in a confined space where ambient air temperatures may rise and prevent adequate cooling.

Water cooled compressors have water jackets around the cylinders, and cooling water is circulated through the jackets. Overcooling is wasteful and costly, and can lead to corrosion and wear within the compressor. Temperature control of the cooling water is important.

Pumped Circulation

Larger single-staged compressors may require a pump to increase the water velocity when thermo-siphon circulation is too slow. The size of the water tank should be discussed with the compressor manufacturer, but in the absence of information, Table 20A can be used as a guide for compressors operating at up to 100 psi.

Single Pass Cooling

The hook-up shown in Fig. II-103 (page 135) is used where water from the local supply is passed directly through the compressor to be cooled. With increasing demands on limited water resources, many water supply authorities do not permit use of water in this way, especially where the warmed water is discharged to waste, and require the use of recirculation systems.

When single pass cooling is used, temperature controls will help ensure consumption is minimized. To avoid the sensor control being in a dead pocket if the control valve ever closes, a small bleed valve is arranged to bypass the control valve. This ensures a small flow past the sensor at all times.

Many compressor manufacturers suggest that the temperature of the water leaving the cylinder jackets should be in the range of 95-120°F. Typical water flow rates needed for compressors are shown in Table 21, but again, these should be checked with the manufacturer where possible.

The supply of cooling water can sometimes be taken from the softened boiler feed water storage tank. The warmed outlet water then becomes a source of pre-heated makeup water for the boiler.

Closed Circuit Cooling

Especially with large compressors, economies are obtained when the cooling water is recycled in a closed circuit. This also minimizes any scaling in the jackets and coolers. The heat may be dissipated at a cooling tower or a mechanical cooler, or sometimes used for space heating in adjacent areas.

Usually with closed circuit cooling, it is preferable to use three-way temperature controls.

Where cooling towers are used, freeze protection of the tower sump may be needed in winter conditions. Often a steam heating coil is installed in the sump with a temperature control set to open when the water temperature falls to say 35°F. A three-way temperature control diverts water direct to the sump instead of to the top of the tower in low temperature conditions. Heat loss from the sump itself then provides sufficient cooling.

Lubricant Coolers

On large reciprocating compressors, and especially on rotary vane compressors, the lubricating oil is usually cooled by passing it through a heat exchanger. Here it gives up heat to cooling water and again the coolant flow should be temperature controlled. See Fig. 65.

Compressor Equipment Guide

Approx. CFM Air = Compressor HP X 5

GPM Cooling Water = $\dfrac{42.5 \times \text{HP/Cylinder}}{8.33 \times \text{Temp. Rise of Cooling Water}}$

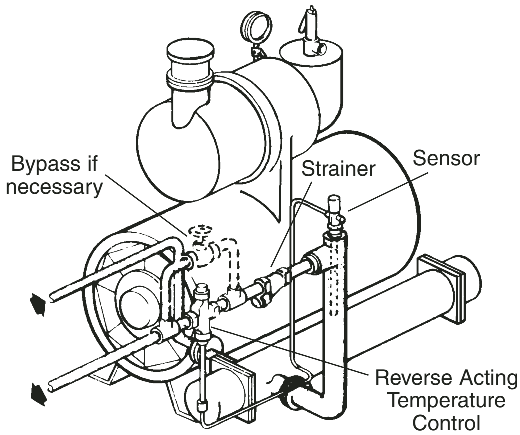


Figure 65
Temperature Control of Water to Oil Cooler

Case in Action: Air System Moisture Separation/Drainage

Air is a vital utility for all process plants, primarily to power control valves, measurement devices and to drive tools, pumps, machinery, etc. Outdoor facilities (i.e. refineries and chemical plants) are faced with continual problems related to water accumulation in the air system.

Free and compressed air carries varying volumes of water at different pressures and temperatures. This affects the amount of entrained water that must be drained in different parts of the system for effective operation.

Desiccant dryers critical for removing water from compressed air distributed to instruments and controls become water-logged due to excess moisture entrained in the free-air supplied. When this occurs, the dryers shut down, curtailing air for distribution.

The heat transfer across the wall of distribution piping creates additional condensing. Moisture, entrained in the air flow, exceeds the capacity of the coalescent filters installed at the point of use. The air-using equipment is then flooded with water, affecting proper operation. This can ruin gauges and instruments and affect control accuracy.

Solution

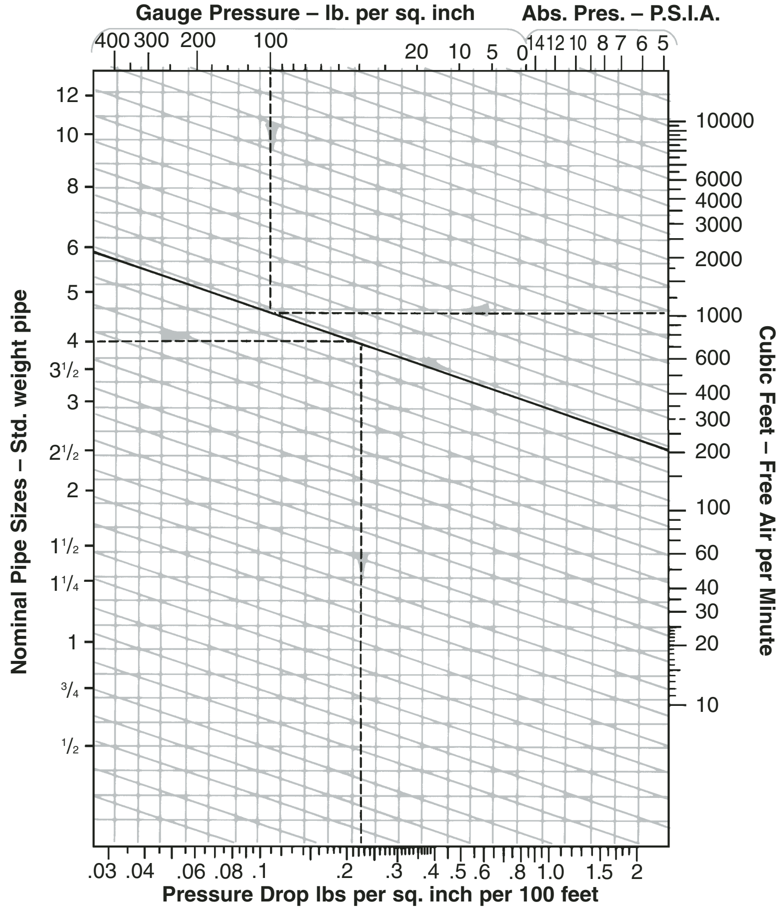
Over 30 separators with drain traps were installed in problem areas providing proper drainage of equipment.

Benefits

- Continuous operation of desiccant dryers, assuring uninterrupted air supply.
- Working with dry air, instrument accuracy is more consistent.
- Damage to gauges and other instruments, caused by entrained water, is prevented, reducing maintenance cost.
- Air hose stations deliver dry air immediately eliminating delay/inconvenience of having to manually drain water from the hoses before use.

Compressed Air Line Pressure Drop

Figure 66: Compressed Air Line Pressure Drop



Pipe Expansion

Table 22: Calculation of Pipe Expansion

Expansion (Δ) = L_0 x Δ_t x a (inches)

L_0 = Length of pipe between anchors (ft)

Δ_t = Temperature difference (°F)

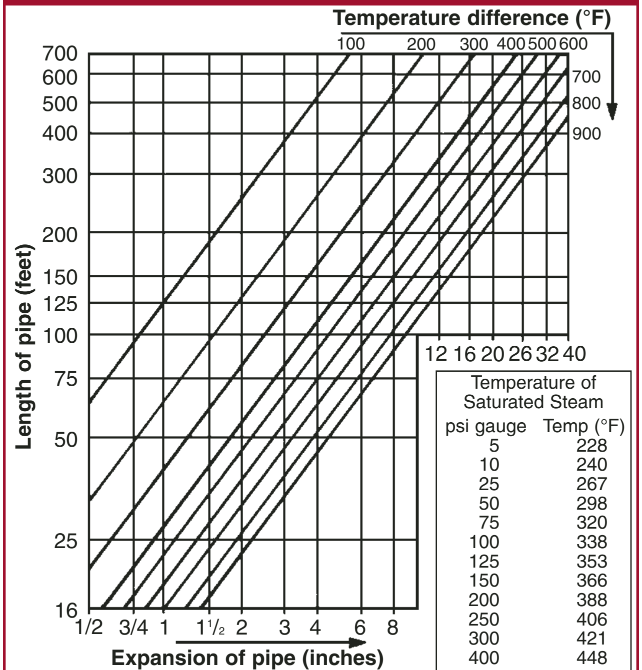
a = Expansion coefficient

For Temp. Range (°F)	30 to 32	32 to 212	32 to 400	32 to 600	32 to 750	32 to 900	32 to 1100	32 to 1300
Mild Steel 0.1-0.2% C	7•1	7•8	8•3	8•7	9•0	9•5	9•7	—
Alloy Steel 1% Cr. 1/2% Mo	7•7	8•0	8•4	8•8	9•2	9•6	9•8	—
Stainless Steel 18% Cr. 8% Ni	10•8	11•1	11•5	11•8	12•1	12•4	12•6	12•8

Expansion Coefficient a x 10^{-5} (inches)

Example 7•1 x 10^{-5} = 0.000071

Figure 67: Expansion Chart for Mild Steam Pipe



Temperature of Saturated Steam

psi gauge	Temp (°F)
5	228
10	240
25	267
50	298
75	320
100	338
125	353
150	366
200	388
250	406
300	421
400	448
500	470

Table 23: Heat Transfer

Average Heat Loss from Oil in Storage Tanks and Pipe Lines

Position	Oil Temperature	Unlagged*	Lagged*
Tank Sheltered	Up to 50°F Up to 80°F Up to 100°F	1.2 1.3 1.4	.3 .325 .35
Tank Exposed	Up to 50°F Up to 80°F Up to 100°F	1.4 1.5 1.6	.35 .375 .4
Tank In Pit	All Temperatures	1.2	—
Pipe Sheltered Line	Up to 80°F 80 to 260°F	1.5 2.3	.375 .575
Pipe Exposed Line	Up to 80°F 80 to 260°F	1.8 2.75	.45 .7

*Heat Transfer Rate in BTU/h ft² °F temperature difference between oil and surrounding air

For rough calculations, it may be taken that 1 ton of fuel oil occupies 36.4 ft³. The specific heat capacity of heavy fuel is 0.45 to 0.48 Btu/lb °F.

Heat Transfer from Steam Coils
Approximately 20 Btu/h ft² of heating surface per °F difference between oil and steam temperature.

Heat Transfer from Hot Water Coils
Approximately 10 Btu/h ft² of heating surface per °F difference between oil and water temperature.

Table 24: Heat Transmission Coefficients

In Btu per sq. ft. per hr. per °F.

Water	Cast Iron	Air or Gas	1.4
Water	Mild Steel	Air or Gas	2.0
Water	Copper	Air or Gas	2.25
Water	Cast Iron	Water	40 to 50
Water	Mild Steel	Water	60 to 70
Water	Copper	Water	62 to 80
Air	Cast Iron	Air	1.0
Air	Mild Steel	Air	1.4
Steam	Cast Iron	Air	2.0
Steam	Mild Steel	Air	2.5
Steam	Copper	Air	3.0
Steam	Cast Iron	Water	160
Steam	Mild Steel	Water	185
Steam	Copper	Water	205
Steam	Stainless Steel	Water	120

The above values are average coefficients for practically still fluids.

The coefficients are dependent on velocities of heating and heated media on type of heating surface, temperature difference and other circumstances. For special cases, see literature, and manufacturer's data.

Table 25: Heat Loss from Open Tanks

Liquid Temp. °F	Heat Loss From Liquid Suface BTU/ft² h			Heat Loss Through Tank Walls BTU/ft² h			
	Evap. Loss	Rad. Loss	Total	Bare Steel	Insulation 1"	2"	3"
90	80	50	130	50	12	6	4
100	160	70	230	70	15	8	6
110	240	90	330	90	19	10	7
120	360	110	470	110	23	12	9
130	480	135	615	135	27	14	10
140	660	160	820	160	31	16	12
150	860	180	1040	180	34	18	13
160	1100	210	1310	210	38	21	15
170	1380	235	1615	235	42	23	16
180	1740	260	2000	260	46	25	17
190	2160	290	2450	290	50	27	19
200	2680	320	3000	320	53	29	20
210	3240	360	3590	360	57	31	22

Table 26: Heat Emission Rates from Pipes Submerged in Water

Published Overall Heat Transfer Rates	Btu/ft² h °F
Tank Coils, Steam/Water (Temperature difference 50°F)	100 to 225
Tank Coils, Steam/Water (Temperature difference 100°F)	175 to 300
Tank Coils, Steam/Water (Temperature difference 200°F)	225 to 475
Reasonable Practical Heat Transfer Rates	
Tank Coils, low pressure with natural circulation	100
Tank Coils, high pressure with natural circulation	200
Tank Coils, low pressure with assisted circulation	200
Tank Coils, high pressure with assisted circulation	300

Table 27: Heat Emission Coefficients from Pipes Submerged in Miscellaneous Fluids

The viscosity of fluids has a considerable bearing on heat transfer characteristics and this varies in any case with temperature. The following figures will therefore serve only as a rough guide.

Immersed steam coil, medium pressure, natural convection.

	Btu/ft² h °F difference
Light Oils	30
Heavy Oils	15 to 20
Fats*	5 to 10

Immersed steam coil, medium pressure, forced convection.

	Btu/ft² h °F difference
Light Oils (220 SSU at 100°F)	100
Medium Oils (1100 SSU at 100°F)	60
Heavy Oils (3833 SSU at 100°F)	30

* Certain materials such as tallow and margarine are solid at normal temperatures but have quite low viscosities in the molten state.

Typical Steam Consumption Rates

Table 28: Typical Steam Consumption Rates

	Operating pressure PSIG	Lbs per hr	
		In use	Maximum
BAKERIES			
Dough room trough, 8 ft long	10	4	
Proof boxes, 500 cu ft capacity		7	
Ovens: Peel Or Dutch Type	10		
White bread, 120 sq ft surface		29	
Rye bread, 10 sq ft surface		58	
Master Baker Ovens		29	
Century Reel, w/pb per 100 lb bread		29	
Rotary ovens, per deck		29	
Bennett 400, single deck		44	
Hubbard (any size)		58	
Middleby-Marshall, w/pb		58	
Baker-Perkins travel ovens, long tray (per 100 lbs)		13	
Baker-Perkins travel ovens, short tray (per 100 lbs)		29	
General Electric		20	
Fish Duothermic Rotary, per deck		58	
Revolving ovens: 8-10 bun pan		29	
12-18 bun pan		58	
18-28 bun pan		87	
BOTTLE WASHING	5		
Soft drinks, beer, etc: per 100 bottles/min		310	
Mill quarts, per 100 cases per hr		58	
CANDY and CHOCOLATE	70		
Candy cooking, 30-gal cooker, 1 hour		46	
Chocolate melting, jacketed, 24" dia		29	
Chocolate dip kettles, per 10 sq ft tank surface		29	
Chocolate tempering, top mixing each 20 sq ft active surface		29	
Candy kettle per sq ft of jacket	30		60
Candy kettle per sq ft of jacket	75		100
CREAMERIES and DAIRIES	15-75		
Creamery cans 3 per min			310
Pasteurizer, per 100 gal heated 20 min			232
DISHWASHERS	10-30		
2-Compartment tub type			58
Large conveyor or roller type			58
Autosan, colt, depending on size		29	117
Champion, depending on size		58	310
Hobart Crescent, depending on size		29	186
Fan Spray, depending on size		58	248
Crescent manual steam control	30		
Hobart Model AM-5	10		
Dishwashing machine	15-20	60-70	
HOSPITAL EQUIPMENT	40-50		
Stills, per 100 gal distilled water		102	
Sterilizers, bed pan		3	
Sterilizers, dressing, per 10" length, approx.		7	
Sterilizers, instrument, per 100 cu in approx.		3	
Sterilizers, water, per 10 gal, approx.		6	
Disinfecting Ovens, Double Door:	40-50		
Up to 50 cu ft, per 10 cu ft approx.		29	
50 to 100 cu ft, per 10 cu ft approx.		21	
100 and up, per 10 cu ft, approx.		16	
Sterilizers, Non-Pressure Type			
For bottles or pasteurization	40		
Start with water at 70°F, maintained for 20 minutes at boiling at a depth of 3"		51	69
Instruments and Utensils:			
Start with water at 70°F, boil vigorously for 20 min:	40		
Depth 3-1/2": Size 8 X 9 X 18"		27	27
Depth 3-1/2": Size 9 X 20 X 10"		30	30
Depth 4": Size 10 X 12 X 22"		39	39
Depth 4": Size 12 X 16 X 24"		60	60
Depth 4": Size 10 X 12 X 36"		66	66
Depth 10": Size 16 X 15 X 20"		92	92
Depth 10": Size 20 X 20 X 24"		144	144
LAUNDRY EQUIPMENT	100		
Vacuum stills, per 10 gal		16	
Spotting board, trouser stretcher		29	
Dress finisher, overcoat shaper, each		58	
Jacket finisher, Susie Q, each		44	

Table 28: Typical Steam Consumption Rates

	Operating pressure PSIG	Lbs per hr	
		In use	Maximum
Air vacuum finishing board, 18" Mushroom Topper, ea.		20	
Steam irons, each		4	
Flat Iron Workers:	100		
48" X 120", 1 cylinder		248	
48" X 120", 2 cylinder		310	
4-Roll, 100 to 120"		217	
6-Roll, 100 to 120"		341	
8-Roll, 100 to 120"		465	
Shirt Equipment	100		
Single cuff, neckband, yoke No. 3, each		7	
Double sleeve		13	
Body		29	
Bosom		44	
Dry Rooms	100		
Blanket		20	
Conveyor, per loop, approx.		7	
Truck, per door, approx.		58	
Curtain, 50 X 114		29	
Curtain, 64 X 130		58	
Starch cooker, per 10 gal cap		7	
Starcher, per 10-in. length approx.		5	
Laundry presses per 10-in. length approx.		7	
Handy irons, per 10-in. length, approx.		5	
Collar equipment: Collar and Cuff Ironer		21	
Deodorizer		87	
Wind Whip, single		58	
Wind Whip, double		87	
Tumblers, General Usage Other Source	100		
36", per 10" length, approx.		29	
40", per 10" length, approx.		38	
42", per 10" length, approx.		52	
Vorcone, 46" X 120"		310	
Presses, central vacuum, 42"		20	
Presses, steam, 42"		29	
PLASTIC MOLDING			
Each 12 to 15 sq ft platen surface	125	29	
PAPER MANUFACTURE			
Corrugators per 1,000 sq ft	175	29	
Wood pulp paper, per 100 lb paper	50	372	
RESTAURANT EQUIPMENT	5-20		
Standard steam tables, per ft length		36	
Standard steam tables, per 20 sq ft tank		29	
Bain Marie, per ft length, 30" wide		13	
Bain Marie, per 10 sq ft tank		29	
Coffee urns, per 10 gal, cold make-up		13	
3-compartment egg boiler		13	
Oyster steamers		13	
Clam or lobster steamer		29	
Steam Jacketed Kettles	5-20		
10 gal capacity		13	106
25 gal stock kettle		29	124
40 gal stock kettle		44	140
60 gal stock kettle		58	152
Plate And Dish Warmers	5-20		
Per 100 sq ft shelf		58	
Per 20 cu ft shelf		29	
Warming ovens, per 20 cu ft		29	
Direct vegetable steamer, per compartment		29	80
Potato steamer		29	80
Morandi Proctor, 30 comp., no return		87	
Pot sink, steam jets, average use		29	
Silver burnishers, Tahara		58	
SILVER MIRRORING			
Average steam tables	5	102	
TIRE SHOPS	100		
Truck molds, large		87	
Truck molds, medium		58	
Passenger molds		29	
Section, per section		7	
Puff Irons, each		7	

Specific Heats and Weights

Table 29: Specific Heats and Weights: Various Solids

Material	Specific Gravity	Specific Heat, B.t.u. per Lb. per °F	Material	Specific Gravity	Specific Heat, B.t.u. per Lb. per °F
Aluminum	2.55-2.8	.22	Ice, 32F		.49
Andalusite		.17	Iridium	21.78-22.42	.03
Antimony		.05	Iron, cast	7.03-7.13	.12
Aatite		.20	Iron, wrought	7.6-7.9	.12
Asbestos	2.1-2.8	.20	Labradorite		.19
Augite		.19	Lava		.20
Bakelite, wood filler		.33	Lead	11.34	.03
Bakelite, asbestos filler		.38	Limestone	2.1-2.86	.22
Barite	4.5	.11	Magnetite	3.2	.16
Barium	3.5	.07	Magnesium	1.74	.25
Basalt rock	2.7-3.2	.20	Malachite		.18
Beryl		.20	Manganese	7.42	.11
Bismuth	9.8	.03	Marble	2.6-2.86	.21
Borax	1.7-1.8	.24	Mercury	13.6	.03
Boron	2.32	.31	Mica		.21
Cadmium	8.65	.06	Molybdenum	10.2	.06
Calcite, 32-100F		.19	Nickel	8.9	.11
Calcite, 32-212F		.20	Oligloclose		.21
Calcium	4.58	.15	Orthoclose		.19
Carbon	1.8-2.1	.17	Plaster of Paris		1.14
Carborundum		.16	Platinum	21.45	.03
Cassiterite		.09	Porcelain		.26
Cement, dry		.37	Potassium	0.86	.13
Cement, powder		.2	Pyrexglass		.20
Charcoal		.24	Pyrolusite		.16
Chalcopyrite		.13	Pyroxylin plastics		.34-.38
Chromium	7.1	.12	Quartz, 55-212F	2.5-2.8	.19
Clay	1.8-2.6	.22	Quartz, 32F		.17
Coal	64-.93	.26-.37	Rock salt		.22
Cobalt	8.9	.11	Rubber		.48
Concrete, stone		.19	Sandstone	2.0-2.6	.22
Concrete, cinder		.18	Serpentine	2.7-2.8	.26
Copper	8.8-8.95	.09	Silk		.33
Corundum		.10	Silver	10.4-10.6	.06
Diamond	3.51	.15	Sodium	0.97	.30
Dolomite rock	2.9	.22	Steel	7.8	.12
Fluorite		.22	Stone		.20
Fluorspar		.21	Stoneware		.19
Galena		.05	Talc	2.6-2.8	.21
Garnet		.18	Tar	1.2	.35
Glass, common	2.4-2.8	.20	Tallurium	6.0-6.24	.05
Glass, crystal	2.9-3.0	.12	Tin	7.2-7.5	.05
Glass, plate	2.45-2.72	.12	Tile, hollow		.15
Glass, wool		.16	Titanium	4.5	.14
Gold	19.25-19.35	.03	Topaz		.21
Granite	2.4-2.7	.19	Tungsten	19.22	.04
Hematite	5.2	.16	Vanadium	5.96	.12
Hornblende	3.0	.20	Vulcanite		.33
Hypersthene		.19	Wood	.35-.99	.32-.48
Ice, -112F		.35	Wool	1.32	.33
Ice, -40F		.43	Zinc blend	3.9-4.2	.11
Ice, -4F		.47	Zinc	6.9-7.2	.09

Table 30: Specific Heats and Weights: Various Liquids

Liquid	Specific Gravity	Specific Heat, B.t.u. per Lb. per °F	Liquid	Specific Gravity	Specific Heat, B.t.u. per Lb. per °F
Acetone	0.790	.51	Fuel Oil	.86	.45
Alcohol, ethyl 32°F	0.789	.55	Fuel Oil	.81	.50
Alcohol, ethyl, 105°F	0.789	.65	Gasoline		.53
Alcohol, methyl, 40-50°F	0.796	.59	Glycerine	1.26	.58
Alcohol, methyl, 60-70°F	0.796	.60	Kerosene		.48
Ammonia, 32°F	0.62	1.10	Mercury	13.6	.033
Ammonia, 104°F		1.16	Naphthalene	1.14	.41
Ammonia, 176°F		1.29	Nitrobenzole		.36
Ammonia, 212°F		1.48	Olive Oil	.91-.94	.47
Ammonia, 238°F		1.61	Petroleum		.51
Anilin	1.02	.52	Potassium Hydrate	1.24	.88
Benzol		.42	Sea Water	1.0235	.94
Calcium Chloride	1.20	.73	Sesame Oil		.39
Castor Oil		.43	Sodium Chloride	1.19	.79
Citron Oil		.44	Sodium Hydrate	1.27	.94
Diphenylamine	1.16	.46	Soybean Oil		.47
Ethyl Ether		.53	Toluol	.866	.36
Ethylene Glycol		.53	Turpentine	.87	.41
Fuel Oil	.96	.40	Water	1	1.00
Fuel Oil	.91	.44	Xylene	.861-.881	.41

Table 31: Specific Heats and Weights: Gas and Vapors

Gas or Vapor	Specific Heat, B.t.u. per Lb. per °F at Constant Pressure	Specific Heat, B.t.u. per Lb. per °F at Constant Volume	Gas or Vapor	Specific Heat, B.t.u. per Lb. per °F at Constant Pressure	Specific Heat, B.t.u. per Lb. per °F at Constant Volume
Acetone	.35	.315	Ether	.48	.466
Air, dry, 50°F	.24	.172	Hydrochloric acid	.19	.136
Air, dry, 32-392°F	.24	.173	Hydrogen	3.41	2.410
Air, dry, 68-824°F	.25	.178	Hydrogen sulphide	.25	.189
Air, dry 68-1166°F	.25	.184	Methane	.59	.446
Air, dry, 68-1472°F	.26	.188	Nitrogen	.24	.170
Alcohol, C_2H_5OH	.45	.398	Nitric oxide	.23	.166
Alcohol, CH_3OH	.46	.366	Nitrogen tetroxide	1.12	1.098
Ammonia	.54	.422	Nitrous oxide	.21	.166
Argon	.12	.072	Oxygen	.22	.157
Benzene, C_6H_6	.26	.236	Steam, 1 psia 120-600 °F	.46	.349
Bromine	.06	.047	Steam, 14.7 psia 220-600 °F	.47	.359
Carbon dioxide	.20	.150	Steam, 150 psia 360-600 °F	.54	.421
Carbon monoxide	.24	.172	Sulphur dioxide	.15	.119
Carbon disulphide	.16	.132			
Chlorine	.11	.082			
Chloroform	.15	.131			

Specific Heats and Weights

Table 32: Specific Heats and Weights: Foodstuffs

Food	Specific Heat, B.t.u. per Lb. per °F above freezing	Specific Heat, B.t.u. per Lb. per °F below freezing
Apples	.87	.42
Apricots, fresh	.88	.43
Artichokes	.87	.42
Asparagus	.94	.45
Asparagus beans	.88	.43
Avocados	.72	.37
Bananas	.80	.40
Barracuda	.80	.40
Bass	.82	.41
Beef, carcass	.68	.48
Beef, flank	.56	.32
Beef, Loin	.66	.35
Beef, rib	.67	.36
Beef, round	.74	.38
Beef, rump	.62	.34
Beef, shanks	.76	.39
Beef, corned	.63	.34
Beets	.90	.43
Blackberries	.87	.42
Blueberries	.87	.42
Brains	.84	.41
Broccoli	.92	.44
Brussels sprouts	.88	.43
Butter	.30	.24
Butterfish	.77	.39
Cabbage	.94	.45
Carp	.82	.41
Carrots	.91	.44
Cauliflower	.93	.44
Celery	.94	.45
Chard	.93	.44
Cherries, sour	.88	.43
Cherries, sweet	.84	.41
Chicken, squab	.80	.40
Chicken, broilers	.77	.39
Chicken, fryers	.74	.38
Chicken, hens	.65	.35
Chicken, capons	.88	.44
Clams, meat only	.84	.41
Coconut, meat and milk	.68	.36
Coconut, milk only	.95	.45
Codfish	.86	.42
Cod Roe	.76	.39
Corn	.84	.423
Cowpeas, fresh	.73	.39
Cowpeas, dry	.28	.22
Crabs	.84	.41
Crab apples	.85	.41
Cranberries	.90	.43
Cream	.90	.38
Cucumber	.98	.45
Currants	.97	.45
Dandelion greens	.88	.43
Dates	.20	.007
Eels	.77	.39
Eggs	.76	.40
Eggplant	.94	.45
Endive	.95	.45
Figs, fresh	.82	.41
Figs, dried	.39	.26
Figs, candied	.37	.26
Flounders	.86	.42
Flour	.38	.28
Frogs legs	.88	.44
Garlic	.79	.40
Gizzards	.78	.39
Goose	.61	.34
Gooseberry	.86	.42
Granadilla	.84	.41
Grapefruit	.91	.44
Grapes	.86	.42
Grape juice	.82	.41
Guavas	.86	.42
Guinea hen	.75	.38
Haddoock	.85	.42
Halibut	.80	.40
Herring, smokes	.71	.37
Horseradish, fresh	.79	.40
Horseradish, prepared	.88	.43
Ice Cream	.74	.40
Kale	.89	.43
Kidneys	.81	.40
Kidney beans, dried	.28	.23
Kohlrabi	.92	.44
Kumquats	.85	.41
Lamb, carcass	.73	.38
Lamb, leg	.71	.37
Lamb, rib cut	.61	.34
Lamb, shoulder	.67	.35
Lard	.54	.31
Leeks	.91	.44
Lemons	.91	.44
Lemon joice	.92	.44
Lettuce	.96	.45
Lima beans	.73	.38
Limes	.89	.43
Lime juice	.93	.44
Litchi fruits, dried	.39	.26
Lobsters	.82	.41
Loganberries	.86	.42
Loganberry joice	.91	.44
Milk, cow	.90	.47
Mushrooms, fresh	.93	.44
Mushrooms, dried	.30	.23
Muskmelons	.94	.45
Nectarines	.86	.42
Nuts	.28	.24
Olives, green	.80	.40
Onions	.90	.43
Onions, Welsh	.91	.44

Table 32: Specific Heats and Weights: Foodstuffs

Food	Specific Heat, B.t.u. per Lb. per °F above freezing	Specific Heat, B.t.u. per Lb. per °F below freezing
Oranges, fresh	.90	.43
Orange juice	.89	.43
Oysters	.84	.41
Peaches, Georgia	.87	.42
Peaches, N. Carolina	.89	.43
Peaches, Maryland	.90	.43
Peaches, New Jersey	.91	.44
Peach juice, fresh	.89	.43
Pears, Bartlett	.89	.43
Pears, Beurre Bosc	.85	.41
Pears, dried	.39	.26
Peas, young	.85	.41
Peas, medium	.81	.41
Peas, old	.88	.43
Peas, split	.28	.23
Peppers, ripe	.91	.44
Perch	.82	.41
Persimmins	.72	.37
Pheasant	.75	.36
Pickerel	.84	.41
Pickels, sweet	.82	.41
Pickels, sour and dill	.96	.45
Pickels, sweet mixed	.78	.29
Pickels, sour mixed	.95	.45
Pig's feet, pickled	.50	.31
Pike	.84	.41
Pineapple, fresh	.88	.43
Pineapple, sliced or crushed	.82	.41
Pineapple joice	.90	.43
Plums	.89	.43
Pomegranate	.85	.41
Pompano	.77	.39
Porgy	.81	.40
Pork, bacon	.36	.25
Pork, ham	.62	.34
Pork, loin	.66	.35
Pork, shoulder	.59	.33
Pork, spareribs	.62	.34
Pork, smoked ham	.65	.35
Pork, salted	.31	.24
Potatoes	.82	.41
Prickly pears	.91	.43
Prunes	.81	.40
Pumpkin	.92	.44
Quinces	.88	.43
Rabbit	.76	.39
Radishes	.95	.45
Raisins	.39	.26
Raspberries, black	.85	.41
Raspberries, red	.89	.43
Raspberry juice, black	.91	.44
Raspberry juice, red	.93	.44
Reindeer	.73	.37
Rhubarb	.96	.45
Rose Apple	.89	.43
Rutabagas	.91	.44
Salmon	.71	.37
Sand dab, California	.86	.42
Sapodilla	.91	.44
Sapote	.73	.37
Sauerkraut	.93	.44
Sausage, beef and pork	.56	.32
Sausage, bockwurst	.71	.37
Sausage, bologna	.71	.37
Sausage, frankfurt	.69	.36
Sausage, salami	.45	.28
Sardines	.77	.39
Shad	.76	.39
Shrimp	.83	.41
Spanish mackerel	.73	.39
Shad	.76	.39
Shrimp	.83	.41
Spanish mackerel	.73	.39
Strawberries	.95	.45
Strawberry juice	.79	.39
String beans	.91	.44
Sturgeon, raw	.83	.41
Sturgeon, smoked	.71	.37
Sugar apple, fresh	.79	.39
Sweet potatoes	.75	.38
Swordfish	.80	.40
Terrapin	.80	.40
Tomatoes, red	.95	.45
Tomatoes, green	.96	.45
Tomato juice	.95	.45
Tongue, beef	.74	.38
Tongue, calf	.79	.40
Tongue, lamb	.76	.38
Tongue, pork	.74	.39
Tongue, sheep	.69	.36
Tripe, beef	.83	.41
Tripe, pickled	.89	.43
Trout	.82	.41
Tuna	.76	.39
Turkey	.67	.35
Turnips	.93	.44
Turtle	.84	.41
Veal, carcass	.74	.38
Veal, flank	.65	.35
Veal, loin	.75	.38
Veal, rib	.73	.37
Veal, shank	.77	.39
Veal, quarter	.74	.38
Venison	.78	.39
Watercress	.95	.45
Watermelons	.94	.45
Whitefish	.76	.39
Yams	.78	.39

Conversions

Table 33: Conversions

To Convert	Into	Multiply by
A		
Acres	sq. feet	43,560.0
Atmospheres	cms. of mercury	76.0
Atmospheres	ft. of water (at 4°C)	33.90
Atmospheres	in. of mercury (at 0°C)	29.92
Atmospheres	kgs./sq. cm.	1.0333
Atmospheres	pounds/sq. in.	14.70
B		
Barrels (U.S. liquid)	gallons	31.5
Barrels (oil)	gallons (oil)	42.0
Btu	foot-lbs	778.3
Btu	grams-calories	252.0
Btu	horsepower-hrs.	3.931×10^{-4}
Btu	kilowatt-hrs	2.928×10^{-4}
Btu/hr.	horsepower	3.931×10^{-4}
Btu/hr.	watts	0.2931
C		
Calories, gram (mean)	B.t.u. (mean)	3.9685×10^{-3}
Centigrade	Fahrenheit	9/5(C° + 40) -40
Centimeters	feet	3.281×10^{-2}
Centimeters	inches	0.3937
Centimeters	mils	393.7
Centimeters of mercury	atmospheres	0.01316
Centimeters of mercury	feet of water	0.4461
Centimeters of mercury	pounds/sq. in.	0.1934
Circumference	radians	6.283
Cubic centimeters	cu. feet	3.531×10^{-5}
Cubic centimeters	cu. inches	0.06102
Cubic centimeters	gallons (U.S. liq.)	2.642×10^{-4}
Cubic feet	cu. cms.	28,320.0
Cubic feet	cu. inches	1,728.0
Cubic feet	gallons (U.S. liq.)	7.481
Cubic feet	liters	28.32
Cubic feet	quarts (U.S. liq.)	29.92
Cubic feet/min.	gallons/sec.	0.1247
Cubic feet/min.	pounds of water/min.	62.43
Cubic inches	cu. cms.	16.39
Cubic inches	gallons	4.329×10^{-3}
Cubic inches	quarts (U.S. liq.)	0.01732
Cubic meters	cu. feet	35.31
Cubic meters	gallons (U.S. liq)	264.2
Cubic yards	cu. feet	27.0
Cubic yards	cu. meters	0.7646
Cubic yards	gallons (U.S. liq.)	202.0
D		
Degrees (angle)	radians	0.01745
Drams (apothecaries' or troy)	ounces (avoirdupois)	0.13714
Drams (apothecaries' or troy)	ounces (troy)	0.125
Drams (U.S. fluid or apothecary)	cubic cm.	3.6967
Drams	grams	1.772
Drams	grains	27.3437
Drams	ounces	0.0625
F		
Fahrenheit	centigrade	5/9(F + 40) - 40
Feet	centimeters	30.48
Feet	kilometers	3.048×10^{-4}
Feet	meters	0.3048
Feet	miles (naut.)	1.645×10^{-4}
Feet	miles (stat.)	1.894×10^{-4}
Feet of water	atmospheres	0.02950
Feet of water	in. of mercury	0.8826
Feet of water	kgs./sq. cm.	0.03045
Feet of water	kgs./sq. meter	304.8
Feet of water	pounds/sq. ft.	62.43
Feet of water	pounds/sq. in.	0.4335
Foot-pounds	Btu	1.286×10^{-3}
Foot-pounds	gram-calories	0.3238
Foot-pounds	hp.-hrs.	5.050×10^{-7}
Foot-pounds	kilowatt-hrs.	3.766×10^{-7}
Foot-pounds/min.	Btu/min.	1.286×10^{-8}
Foot-pounds/min.	horsepower	3.030×10^{-5}
Foot-pounds/sec.	Btu/hr.	4.6263
Furlongs	miles	0.125
Furlongs	feet	660.0
G		
Gallons	cu. cms.	3,785.0
Gallons	cu. feet	0.1337
Gallons	cu. inches	231.0
Gallons	cu. meters	3.785×10^{-3}
Gallons	cu. yards	4.951×10^{-3}
Gallons	liters	3.785
Gallons (liq. Br. Imp.)	gallons (U.S. liq.)	1.20095
Gallons (U.S.)	gallons (Imp.)	0.83267
Gallons of water	pounds of water	8.3453
Gallons/min.	cu. ft./sec.	2.228×10^{-3}
Gallons/min.	liters/sec.	0.06308
Gallons/min.	cu. ft./hr.	8.0208
Grains (troy)	grains (avdp.)	1.0
Grains (troy)	grams	0.06480
Grains (troy)	ounces (avdp.)	0.286×10^{-3}
Grains (troy)	pennyweight (troy)	0.04167
Grains/U.S. gal.	parts/million	17.118
Grains/U.S. gal.	pounds/million gal.	142.86
Grains/Imp. gal.	parts/million	14.286
Grams	grains	15.43
Grams	ounces (avdp.)	0.03527
Grams	ounces (troy)	0.03215
Grams	poundals	0.07093
Grams	pounds	2.205×10^{-3}
Grams/liter	parts/million	1,000.0
Gram-calories	Btu	3.9683×10^{-3}
Gram-caloories	foot-pounds	3.0880
Gram-calories	kilowatt-hrs.	1.1630×10^{-4}
Gram-calories	watt-hrs.	1.1630×10^{-3}
H		
Horsepower	Btu/min.	42.40
Horsepower	foot-lbs./min.	33,000.
Horsepower	foot-lbs./sec.	550.0
Horsepower (metric) (542.5 ft. lb./sec.)	horsepower (550 ft. lb./sec.)	0.9863
Horsepower (550 ft. lb./sec.)	horsepower (metric) (542.5 ft. lb./sec.)	1.014
Horsepower	kilowatts	0.7457
Horsepower	watts	745.7
Horsepower (boiler)	Btu/hr.	33,520.
Horsepower (boiler)	kilowatts	9.803
Horsepower-hrs.	Btu	2,547.
Horsepower-hrs.	foot-lbs.	1.98×10^{6}
Horsepower-hrs.	kilowatt-hrs.	0.7457
I		
Inches	centimeters	2.540
Inches	meters	2.540×10^{-2}
Inches	millimeters	25.40
Inches	yards	2.778×10^{-2}
Inches of mercury	atmospheres	0.03342
Inches of mercury	feet of water	1.133
Inches of mercury	kgs./sq. cm.	0.03453
Inches of mercury	kgs./sq.meter	345.3
Inches of mercury	pounds/sq. ft.	70.73
Inches of mercury	pounds/sq. in.	0.4912
Inches of water (at 4°C)	atmospheres	2.458×10^{-3}
Inches of water (at 4°C)	inches of mercury	0.07355
Inches of water (at 4°C)	kgs./sq./ cm.	2.538×10^{-3}
Inches of water (at 4°C)	ounces/sq. in.	0.5781
Inches of water (at 4°C)	pounds/sq. ft.	5.204
Inches of water (at 4°C)	pounds/sq. in.	0.03613
J		
Joules	Btu	9.480×10^{-4}
K		
Kilograms	grams	1,000.0
Kilograms	pounds	2.205
Kilograms/cu. meter	pounds/cu. ft.	0.06243
Kilograms/cu. meter	pounds/cu. in.	3.613×10^{-5}
Kilograms/sq. cm.	atmospheres	0.9678
Kilograms/sq. cm.	feet of water	32.84

Table 33: Conversions

To Convert	Into	Multiply by
Kilograms/sq. cm.	inches of mercury	28.96
Kilograms/sq. cm.	pounds/sq. ft.	2,048
Kilograms/sq. cm.	pounds/sq. in.	14.22
Kilograms/sq. meter	atmospheres	9.678 x 10^{-5}
Kilograms/sq. meter	feet of water	3.281 x 10^{-3}
Kilograms/sq. meter	inches of mercury	2.896 x 10^{-3}
Kilograms/sq. meter	pounds/sq. ft.	0.2048
Kilograms/sq. meter	pounds/sq. in.	1.422 x 10^{-3}
Kilograms/sq. mm.	kgs./sq. meter	10^{6}
Kilogram-calories	Btu	3.968
Kilogram-calories	foot-pounds	3,088.
Kilogram-calories	hp-hrs.	1.560 x 10^{-3}
Kilogram-calories	kilowatt-hrs.	1.163 x 10^{-3}
Kilogram meters	Btu	9.294 x 10^{-3}
Kilometers	centimeters	10^{5}
Kilometers	feet	3,281.
Kilometers	miles	0.6214
Kilowatts	Btu/min.	56.87
Kilowatts	foot-lbs/min.	4.426 x 10^{4}
Kilowatts	foot-lbs/sec.	737.6
Kilowatts	horsepower	1.341
Kilowatts	watts	1,000.0
Kilowatt-hrs	Btu	3,413.
Kilowatt-hrs.	foot-lbs.	2.655 x 10^{6}
Kilowatt-hrs	horsepower-hrs	1.341
Knots	statute miles/hr.	1.151
	L	
Liters	cu. cm.	1,000.0
Liters	cu. feet	0.03531
Liters	cu. inches	61.02
Liters	gallons (U.S. liq.)	0.2642
	M	
Meters	centimeters	100.0
Meters	feet	3.281
Meters	inches	39.37
Meters	millimeters	1,000.0
Meters	yards	1.094
Microns	inches	39.37 x 10^{-6}
Microns	meters	1 x 10^{-6}
Miles (statute)	feet	5.280.
Miles (statute)	kilometers	1.609
Miles/hr.	cms./sec.	44.70
Miles/hr.	feet/min.	88.
Mils	inches	0.001
Mils	yards	2.778 x 10^{-5}
	N	
Nepers	decibels	8.686
	O	
Ohms	megohms	10^{-6}
Ohms	microhms	10^{6}
Ounces (avoirdupois)	drams	16.0
Ounces (avoirdupois)	grains	437.5
Ounces (avoirdupois	grams	28.35
Ounces (avoirdupois)	pounds	0.0625
Ounces (avoirdupois	ounces (troy)	0.9115
Ounces (troy)	grains	480.0
Ounces (troy)	grains	31.10
Ounces (troy)	ounces (avdp.)	1.09714
Ounces (troy)	pounds (troy)	0.08333
	P	
Parts/million	grains/U.S. gal.	0.0584
Parts/million	grains/Imp. gal	0.07016
Parts/million	pounds/million gal.	8.33
Pounds (avoirdupois)	ounces (troy)	14.58
Pounds (avoirdupois)	drams	256.
Pounds (avoirdupois)	grains	7,000.
Pounds (avoirdupois)	grams	453.59
Pounds (avoirdupois)	kilograms	0.454
Pounds (avoirdupois)	ounces	16.0
Pounds (avoirdupois)	tons (short)	0.0005
Pounds (troy)	ounces (avdp.)	13.1657
Pounds of water	cu. feet	0.01602
Pounds of water	cu. inches	27.68
Pounds of water	gallons	0.1198
Pounds of water/min.	cu. ft/sec.	2.670 x 10^{-4}
Pounds/cu. ft.	grams/cu. cm.	0.01602
Pounds/cu. ft.	kgs./cu. meter	16.02
Pounds/cu. ft.	pounds/cu. in.	5.787 x 10^{-4}
Pounds/cu. in.	pounds/cu. ft.	1,728.
Pounds/sq. ft.	atmospheres	4.725 x 10^{-4}
Pounds/sq. ft.	feet of water	0.01602
Pounds/sq. ft.	inches of mercury	0.01414
Pounds/sq. in.	atmospheres	0.06804
Pounds/sq. in.	feet of water	2.307
Pounds/sq. in.	inches of mercury	2.036
Pounds/sq. in.	kgs./sq. meter	703.1
Pounds/sq. in.	pounds/sq. ft.	144.0
	R	
Radians	degrees	57.30
Revolutions/min.	degrees/sec.	6.0
Revolutions/min.	radians/sec.	0.1047
Revolutions/min.	revs./sec.	0.01667
	S	
Square centimeters	sq. feet	1.076 x 10^{-3}
Square centimeters	sq. inches	0.1550
Square centimeters	sq. meters	0.0001
Square centimeters	sq. millimeters	100.0
Square feet	acres	2.296 x 10^{-5}
Square feet	sq. cms.	929.0
Square feet	sq. inches	144.0
Square feet	sq. miles	3.587 x 10^{-6}
Square inches	sq. cms.	6.452
Square inches	sq. feet	6.944 x 10^{-3}
Square inches	sq. yards	7.716 x 10^{-4}
Square meters	sq. feet	10.76
Square meters	sq. inches	1,550.
Square meters	sq. millimeters	10^{6}
Square meters	sq. yards	1.196
Square millimeters	sq. inches	1.550 x 10^{-3}
Square yards	sq. feet	9.0
Square yards	sq. inches	1,296.
Square yards	sq. meters	0.8361
	T	
Temperature (°C) + 273	absolute temperature (°C)	1.0
Temperature (°C) + 17.78	temperature (°F)	1.8
Temperature (°F) + 460	absolute temperature (°F)	1.0
Temperature (°F) - 32	temperature (°C)	5/9
Tons (long)	kilograms	1,016.
Tons (long)	pounds	2,240.
Tons (long)	tons (short)	1.120
Tons (metric)	kilograms	1,000.
Tons (metric)	pounds	2,205.
Tons (short)	kilograms	907.2
Tons (short)	pounds	2,000.
Tons (short)	tons (long)	0.89287
Tons of water/24 hrs.	pounds of water/hr.	83.333
Tons of water/24 hrs.	gallons/min.	0.16643
Tons of water/24 hrs.	cu. ft./hr.	1.3349
	W	
Watts	Btu/hr.	3.4129
Watts	Btu/min.	0.05688
Watts	horsepower	1.341 x 10^{-3}
Watts	horsepower (metric)	1.360 x 10^{-3}
Watts	kilowatts	0.001
Watts (Abs.)	B.t.u. (mean)/min.	0.056884
Watt-hours	Btu	3.413
Watt-hours	horsepower-hrs.	1.341 x 10^{-3}
	Y	
Yards	centimeters	91.44
Yards	kilometers	9.144 x 10^{-4}
Yards	meters	0.9144

Flow of Water through Schedule 40 Steel Pipe

Table 34: Flow of Water through Schedule 40 Steel Pipe

Pressure Drop per 1,000 Feet of Schedule 40 Steel Pipe, in pounds per square inch

Dschg Gals. per Min.	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressue Drop	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressure Drop	Vel. Ft. per Sec.	Pressure Drop
	1"																	
1	.37	0.49	1-1/4"															
2	.74	1.70	0.43	.045	1-1/2"													
3	1.12	3.53	0.64	0.94	0.47	0.44												
4	1.49	5.94	0.86	1.55	0.63	0.74	2"											
5	1.86	9.02	1.07	2.36	0.79	1.12												
6	2.24	12.25	1.28	3.30	0.95	1.53	.57	0.46	2-1/2"									
8	2.98	21.1	1.72	5.52	1.26	2.63	.76	.075										
10	3.72	30.8	2.14	8.34	1.57	3.86	.96	1.14	.67	0.48								
15	5.60	64.6	3.21	17.6	2.36	8.13	1.43	2.33	1.00	0.99	3"		3-1/2"					
20	7.44	110.5	4.29	29.1	3.15	13.5	1.91	3.86	1.34	1.64	.87	0.59						
25			5.36	43.7	3.94	20.2	2.39	5.81	1.68	2.48	1.08	0.67	.81	0.42	4"			
30			6.43	62.9	4.72	29.1	2.87	8.04	2.01	3.43	1.30	1.21	.97	0.60				
35			7.51	82.5	5.51	38.2	3.35	10.95	2.35	4.49	1.52	1.58	1.14	0.79	.88	0.42		
40					6.30	47.8	3.82	13.7	2.68	5.88	1.74	2.06	1.30	1.00	1.01	0.53		
45					7.08	60.6	4.30	17.4	3.00	7.14	1.95	2.51	1.46	1.21	1.13	0.67		
50					7.87	74.7	4.78	20.6	3.35	8.82	2.17	3.10	1.62	1.44	1.26	0.80	5"	
60							5.74	29.6	4.02	12.2	2.60	4.29	1.95	2.07	1.51	1.10		
70							6.69	38.6	4.69	15.3	3.04	5.84	2.27	2.71	1.76	1.50	1.12	0.48
80	6"						7.65	50.3	5.37	21.7	3.48	7.62	2.59	3.53	2.01	1.87	1.28	0.63
90							8.60	63.6	6.04	26.1	3.91	9.22	2.92	4.46	2.26	2.37	1.44	0.80
100	1.11	0.39					9.56	75.1	6.71	32.3	4.34	11.4	3.24	5.27	2.52	2.81	1.60	0.95
125	1.39	0.56							8.38	48.2	5.42	17.1	4.05	7.86	3.15	4.38	2.00	1.48
150	1.67	0.78							10.06	60.4	6.51	23.5	4.86	11.3	3.78	6.02	2.41	2.04
175	1.94	1.06	8"						11.73	90.0	7.59	32.0	5.67	14.7	4.41	8.20	2.81	2.78
200	2.22	1.32									8.68	39.7	6.48	19.2	5.04	10.2	3.21	3.46
225	2.50	1.66	1.44	0.44							9.77	50.2	7.29	23.1	5.67	12.9	3.61	4.37
250	2.78	2.05	1.60	0.55							10.85	61.9	8.10	28.5	6.30	15.9	4.01	5.14
275	3.06	2.36	1.76	0.63							11.94	75.0	8.91	34.4	6.93	18.3	4.41	6.22
300	3.33	2.80	1.92	0.75							13.02	84.7	9.72	40.9	7.56	21.8	4.81	7.41
325	3.61	3.29	2.08	0.88									10.53	45.5	8.18	25.5	5.21	8.25
350	3.89	3.62	2.24	0.97									11.35	52.7	8.82	29.7	5.61	9.57
375	4.16	4.16	2.40	1.11									12.17	60.7	9.45	32.3	6.01	11.0
400	4.44	4.72	2.56	1.27									12.97	68.9	10.08	36.7	6.41	12.5
425	4.72	5.34	2.72	1.43	10"								13.78	77.8	10.70	41.5	6.82	14.1
450	5.00	5.96	2.88	1.60									14.59	87.3	11.33	46.5	7.22	15.0
475	5.27	6.66	3.04	1.69	1.93	0.30									11.96	51.7	7.62	16.7
500	5.55	7.39	3.20	1.87	2.04	0.63									12.59	57.3	8.02	18.5
550	6.11	8.94	3.53	2.26	2.24	0.70									13.84	69.3	8.82	22.4
600	6.66	10.6	3.85	2.70	2.44	0.86	12"								15.10	82.5	9.62	26.7
650	7.21	11.8	4.17	3.16	2.65	1.01											10.42	31.3
700	7.77	13.7	4.49	3.69	2.85	1.18	2.01	0.48									11.22	36.3
750	8.32	15.7	4.81	4.21	3.05	1.35	2.15	0.55	14"								12.02	41.6
800	8.88	17.8	5.13	4.79	3.26	1.54	2.29	0.62									12.82	44.7
850	9.44	20.2	5.45	5.11	3.46	1.74	2.44	0.70	2.02	0.43							13.62	50.5
900	10.00	22.6	5.77	5.73	3.66	1.94	2.58	0.79	2.14	0.48							14.42	56.6
950	10.55	23.7	6.09	6.38	3.87	2.23	2.72	0.88	2.25	0.53							15.22	63.1
1,000	11.10	26.3	6.41	7.08	4.07	2.40	2.87	0.98	2.38	0.59	16"						16.02	70.0
1,100	12.22	31.8	7.05	8.56	4.48	2.74	3.16	1.18	2.61	0.68							17.63	84.6
1,200	13.32	37.8	7.69	10.2	4.88	3.27	3.45	1.40	2.85	0.81	2.18	0.40						
1,300	14.43	44.4	8.33	11.3	5.29	3.86	3.73	1.56	3.09	0.95	2.36	0.47						
1,400	15.54	51.5	8.97	13.0	5.70	4.44	4.02	1.80	3.32	1.10	2.54	0.54						
1,500	16.65	55.5	9.62	15.0	6.10	5.11	4.30	2.07	3.55	1.19	2.73	0.62	18"					
1,600	17.76	63.1	10.26	17.0	6.51	5.46	4.59	2.36	3.80	1.35	2.91	0.71						
1,800	19.98	79.8	11.54	21.6	7.32	6.91	5.16	2.98	4.27	1.71	3.27	0.85	2.58	0.48				
2,000	22.20	98.5	12.83	25.0	8.13	8.54	5.73	3.47	4.74	2.11	3.63	1.05	2.88	0.56				
2,500			16.03	39.0	10.18	12.5	7.17	5.41	5.92	3.09	4.54	1.63	3.59	0.88	20"			
3,000			19.24	52.4	12.21	18.0	8.60	7.31	7.12	4.45	5.45	2.21	4.31	1.27	3.45	0.73	24"	
3,500			22.43	71.4	14.25	22.9	10.03	9.95	8.32	6.18	6.35	3.00	5.03	1.52	4.03	0.94		
4,000			25.65	93.3	16.28	29.9	11.48	13.0	9.49	7.92	7.25	3.92	5.74	2.12	4.61	1.22	3.19	0.51
4,500					18.31	37.8	12.90	15.4	10.67	9.36	8.17	4.97	6.47	2.50	5.19	1.55	3.59	0.60
5,000					20.35	46.7	14.34	18.9	11.84	11.6	9.08	5.72	7.17	3.08	5.76	1.78	3.99	0.74
6,000					24.42	67.2	17.21	27.3	14.32	15.4	10.88	8.24	8.62	4.45	6.92	2.57	4.80	1.00
7,000					28.50	85.1	20.08	37.2	16.60	21.0	12.69	12.2	10.04	6.06	8.06	3.50	5.68	1.36
8,000							22.95	45.1	18.98	27.4	14.52	13.6	11.48	7.34	9.23	4.57	6.38	1.78
9,000							25.80	57.0	21.35	34.7	16.32	17.2	12.92	9.20	10.37	5.36	7.19	2.25
10,000							28.63	70.4	23.75	42.9	18.16	21.2	14.37	11.5	11.53	6.63	7.96	2.78
12,000							34.38	93.6	28.50	61.8	21.80	30.9	17.23	16.5	13.83	9.54	9.57	3.71
14,000									33.20	84.0	25.42	41.6	20.10	20.7	16.14	12.0	11.18	5.05
16,000											29.05	54.4	22.96	27.1	18.43	15.7	12.77	6.60

Friction Loss for Water in Feet per 100 ft. Schedule 40 Steel Pipe

Table 35: Friction Loss* for Water in Feet per 100 ft. Schedule 40 Steel Pipe

U.S. Gal/Min.	Velocity Ft/Sec.	hf Friction
3/8" PIPE		
1.4	2.35	9.03
1.6	2.68	11.6
1.8	3.02	14.3
2.0	3.36	17.3
2.5	4.20	26.0
3.0	5.04	36.0
3.5	5.88	49.0
4.0	6.72	63.2
5.0	8.40	96.1
6	10.08	136
7	11.8	182
8	13.4	236
9	15.1	297
10	16.8	364
1/2" PIPE		
2	2.11	5.50
2.5	2.64	8.24
3	3.17	11.5
3.5	3.70	15.3
4.0	4.22	19.7
5	5.28	29.7
6	6.34	42.0
7	7.39	56.0
8	8.45	72.1
9	9.50	90.1
10	10.56	110.6
12	12.7	156
14	14.8	211
16	16.9	270
3/4" PIPE		
4.0	2.41	4.85
5	3.01	7.27
6	3.61	10.2
7	4.21	13.6
8	4.81	17.3
9	5.42	21.6
10	6.02	26.5
12	7.22	37.5
14	8.42	50.0
16	9.63	64.8
18	10.8	80.9
20	12.0	99.0
22	13.2	120
24	14.4	141
26	15.6	165
28	16.8	189
1" PIPE		
6	2.23	3.16
8	2.97	5.20
10	3.71	7.90
12	4.45	11.1
14	5.20	14.7
16	5.94	19.0
18	6.68	23.7
20	7.42	28.9
22	8.17	34.8
24	8.91	41.0
26	9.65	47.8
28	10.39	55.1
30	11.1	62.9
35	13.0	84.4
40	14.8	109
45	16.7	137
50	18.6	168
1-1/4" PIPE		
12	2.57	2.85
14	3.00	3.77
16	3.43	4.83
18	3.86	6.00
20	4.29	7.30
22	4.72	8.72
24	5.15	10.27
26	5.58	11.94
28	6.01	13.7
30	6.44	15.6
35	7.51	21.9
40	8.58	27.1
45	9.65	33.8
50	10.7	41.4
55	11.8	49.7
60	12.9	58.6
65	13.9	68.6
70	15.0	79.2
75	16.1	90.6
1-1/2" PIPE		
16	2.52	2.26
18	2.84	2.79
20	3.15	3.38
22	3.47	4.05
24	3.78	4.76
26	4.10	5.54
28	4.41	6.34
30	4.73	7.20
35	5.51	9.63
40	6.30	12.41
45	7.04	15.49
50	7.88	18.9
55	8.67	22.7
60	9.46	26.7
65	10.24	31.2
70	11.03	36.0
75	11.8	41.2
80	12.6	46.6
85	13.4	52.4
90	14.2	58.7
95	15.0	65.0
100	15.8	71.6
2" PIPE		
25	2.39	1.48
30	2.87	2.10
35	3.35	2.79
40	3.82	3.57
45	4.30	4.40
50	4.78	5.37
60	5.74	7.58
70	6.69	10.2
80	7.65	13.1
90	8.60	16.3
100	9.56	20.0
120	11.5	28.5
140	13.4	38.2
160	15.3	49.5
2-1/2" PIPE		
35	2.35	1.15
40	2.68	1.47
45	3.02	1.84
50	3.35	2.23
60	4.02	3.13
70	4.69	4.18
80	5.36	5.36
90	6.03	6.69
100	6.70	8.18
120	8.04	11.5
140	9.38	15.5
160	10.7	20.0
180	12.1	25.2
200	13.4	30.7
220	14.7	37.1
240	16.1	43.8
3" PIPE		
50	2.17	.762
60	2.60	1.06
70	3.04	1.40
80	3.47	1.81
90	3.91	2.26
100	4.34	2.75
120	5.21	3.88
140	6.08	5.19
160	6.94	6.68
180	7.81	8.38
200	8.68	10.2
220	9.55	12.3
240	10.4	14.5
260	11.3	16.9
280	12.2	19.5
300	13.0	22.1
350	15.2	30
4" PIPE		
100	2.52	.718
120	3.02	1.01
140	3.53	1.35
160	4.03	1.71
180	4.54	2.14
200	5.04	2.61
220	5.54	3.13
240	6.05	3.70
260	6.55	4.30
280	7.06	4.95
300	7.56	5.63
350	8.82	7.54
400	10.10	9.75
450	11.4	12.3
500	12.6	14.4
550	13.9	18.1
600	15.1	21.4
5" PIPE		
160	2.57	.557
180	2.89	.698
200	3.21	.847
220	3.53	1.01
240	3.85	1.19
260	4.17	1.38
300	4.81	1.82
350	5.61	2.43
400	6.41	3.13
450	7.22	3.92
500	8.02	4.79
600	9.62	6.77
700	11.2	9.13
800	12.8	11.8
900	14.4	14.8
1000	16.0	18.2
6" PIPE		
220	2.44	.411
240	2.66	.482
260	2.89	.560
300	3.33	.733
350	3.89	.980
400	4.44	1.25
450	5.00	1.56
500	5.55	1.91
600	6.66	2.69
700	7.77	3.60
800	8.88	4.64
900	9.99	5.81
1000	11.1	7.10
1100	12.2	8.52
1200	13.3	10.1
1300	14.4	11.7
1400	15.5	13.6

* Aging Factor Included

Figure 68: Ashrae Psychrometric Chart No. 1

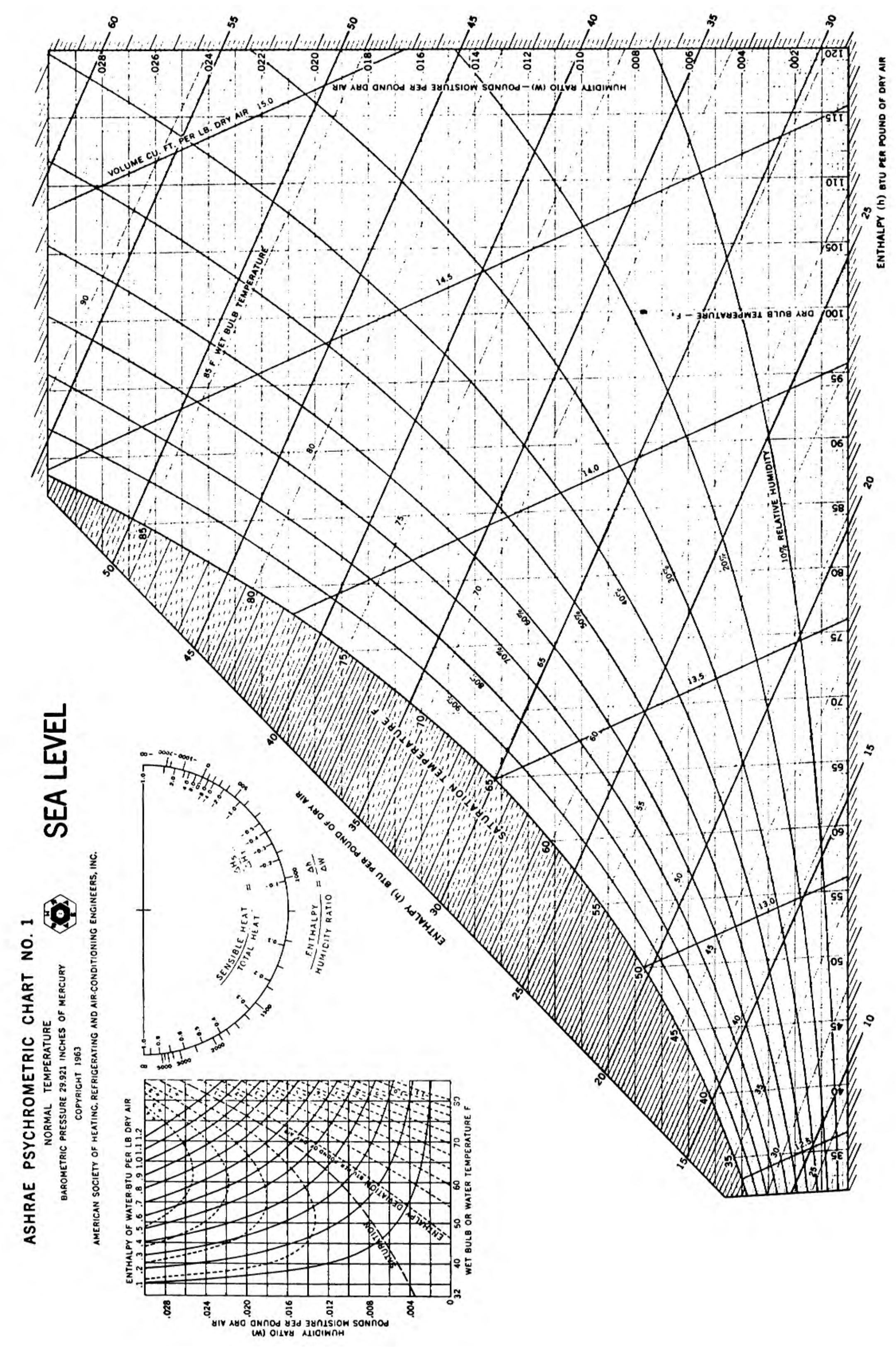


Table 36: Equivalent Length in Feet of New Straight Pipe for Valves and Fittings for Turbulent Flow Only

Fittings			Pipe Size 1/4	3/8	1/2	3/4	1	1¼	1½	2	2½	3	4	5	6	8	10	12	14	16	18	20	24
Regular 90° ELL	Screwed	Steel	2.3	3.1	3.6	4.4	5.2	6.6	7.4	8.5	9.3	11	13										
		C.I.										9.0	11										
	Flanged	Steel			.92	1.2	1.6	2.1	2.4	3.1	3.6	4.4	5.9	7.3	8.9	12	14	17	18	21	23	25	30
		C.I.										3.6	4.8		7.2	9.8	12	15	17	19	22	24	28
Long Radius 90° ELL	Screwed	Steel	1.5	2.0	2.2	2.3	2.7	3.2	3.4	3.6	3.6	4.0	4.6										
		C.I.										3.3	3.7										
	Flanged	Steel			1.1	1.3	1.6	2.0	2.3	2.7	2.9	3.4	4.2	5.0	5.7	7.0	8.0	9.0	9.4	10	11	12	14
		C.I.										2.8	3.4		4.7	5.7	6.8	7.8	8.6	9.6	11	11	13
Regular 45° ELL	Screwed	Steel	.34	.52	.71	.92	1.3	1.7	2.1	2.7	3.2	4.0	5.5										
		C.I.										3.3	4.5										
	Flanged	Steel			.45	.59	.81	1.1	1.3	1.7	2.0	2.6	3.5	4.5	5.6	7.7	9.0	11	13	15	16	18	22
		C.I.										2.1	2.9		4.5	6.3	8.1	9.7	12	13	15	17	20
Tee- Line Flow	Screwed	Steel	.79	1.2	1.7	2.4	3.2	4.6	5.6	7.7	9.3	12	17										
		C.I.										9.9	14										
	Flanged	Steel			.69	.82	1.0	1.3	1.5	1.8	1.9	2.2	2.8	3.3	3.8	4.7	5.2	6.0	6.4	7.2	7.6	8.2	9.6
		C.I.										1.9	2.2		3.1	3.9	4.6	5.2	5.9	6.5	7.2	7.7	8.8
Tee- Branch Flow	Screwed	Steel	2.4	3.5	4.2	5.3	6.6	8.7	9.9	12	13	17	21										
		C.I.										14	17										
	Flanged	Steel			2.0	2.6	3.3	4.4	5.2	6.6	7.5	9.4	12	15	18	24	30	34	37	43	47	52	62
		C.I.										7.7	10		15	20	25	30	35	39	44	49	57
180° Return Bend	Screwed	Steel	2.3	3.1	3.6	4.4	5.2	6.6	7.4	8.5	9.3	11	13										
		C.I.										9.0	11										
	Reg. Flanged	Steel			.92	1.2	1.6	2.1	2.4	3.1	3.6	4.4	5.9	7.3	8.9	12	14	17	19	21	23	25	30
		C.I.										3.6	4.8		7.2	9.8	12	15	17	19	22	24	28
	Long Rad Flanged	Steel			1.1	1.3	1..6	2.0	2.3	2.7	2.9	3.4	4.2	5.0	5.7	7.0	8.0	9.0	9.4	10	11	12	14
		C.I.										2.8	3.4		4.7	5.7	6.8	7.8	8.6	9.6	11	11	13
Globe Valve	Screwed	Steel	21	22	22	24	29	37	42	54	62	79	110										
		C.I.										65	86										
	Flanged	Steel			38	40	45	54	59	70	77	94	120	150	190	260	310	390					
		C.I.										77	99		150	210	270	330					
Gate Valve	Screwed	Steel	.32	.45	.56	.67	.84	1.1	1.2	1.5	1.7	1.9	2.5										
		C.I.										1.6	2.0										
	Flanged	Steel								2.6	2.7	2.8	2.9	3.1	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2
		C.I.										2.3	2.4		2.6	2.7	2.8	2.9	2.9	3.0	3.0	3.0	3.0
Angle Valve	Screwed	Steel	12.8	15	15	15	17	18	18	18	18	18	18										
		C.I.										15	15										
	Flanged	Steel			15	15	17	18	18	21	22	28	38	50	63	90	120	140	160	190	210	240	300
		C.I.										23	31		52	74	98	120	150	170	200	230	280
Swing Check Valve	Screwed	Steel	7.2	7.3	8.0	8.8	11	13	15	19	22	27	38										
		C.I.										22	31										
	Flanged	Steel			3.8	5.3	7.2	10	12	17	21	27	38	50	63	90	120	140					
		C.I.										22	31		52	74	98	120					
Coupling or Union	Screwed	Steel	.14	.18	.21	.24	.29	.36	.39	.45	.47	.53	.65										
		C.I.										.44	.52										
	Bell Mouth Inlet	Steel	.04	.07	.10	.13	.18	.26	.31	.43	.52	.67	.95	1.3	1.6	2.3	2.9	3.5	4.0	4.7	5.3	6.1	7.6
		C.I.										.55	.77		1.3	1.9	2.4	3.0	3.6	4.3	5.0	5.7	7.0
	Square Mouth Inlet	Steel	.44	.68	.96	1.3	1.8	2.6	3.1	4.3	5.2	6.7	9.5	13	16	23	29	35	40	47	53	61	76
		C.I.										5.5	7.7		13	19	24	30	36	43	50	57	70
	Re-entrant Pipe	Steel	.88	1.4	1.9	2.6	3.6	5.1	6.2	8.5	10	13	19	25	32	45	58	70	80	95	110	120	150
		C.I.										11	15		26	37	49	61	73	86	100	110	140
	Y-Strainer			4.6	5.0	6.6	7.7	18	20	27	29	34	42	53	61								
	Sudden Enlargement		$h = \frac{(V_1 - V_2)^2}{2g}$ FEET OF LIQUID; IF $V_2 = 0$ $h = \frac{V_1^2}{2g}$ FEET OF LIQUID																				

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Table 37: ANSI Flange Standards

All Dimensions are in Inches

Pipe Size	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
125 lb CAST IRON – ANSI															
Diameter of Flange	–	–	4 1/4	4 5/8	5	6	7	7 1/2	8 1/2	9	10	11	13 1/2	16	19
Thickness of Flange (min)[1]	–	–	7/16	1/2	9/16	5/8	11/16	3/4	13/16	15/16	15/16	1	1 1/8	1 3/16	1 1/4
Diameter of Bolt Circle	–	–	3 1/8	3 1/2	3 7/8	4 3/4	5 1/2	6	7	7 1/2	8 1/2	9 1/2	11 3/4	14 1/4	17
Number of Bolts	–	–	4	4	4	4	4	4	8	8	8	8	8	12	12
Diameter of Bolts	–	–	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	7/8	7/8
[1] *125 lb flanges have plain faces.*															
250 lb CAST IRON – ANSI															
Diameter of Flange	–	–	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	17 1/2	20 1/2
Thickness of Flange (min)[2]	–	–	11/16	3/4	13/16	7/8	1	1 1/8	1 3/16	1 1/4	1 3/8	1 7/16	1 5/8	1 7/8	2
Diameter of Raised Face	–	–	2 11/16	3 1/16	3 9/16	4 3/16	4 15/16	5 11/16	6 5/16	6 15/16	8 5/16	9 11/16	11 15/16	14 1/16	16 7/16
Diameter of Bolt Circle	–	–	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	15 1/4	17 3/4
Number of Bolts	–	–	4	4	4	8	8	8	8	8	8	12	12	16	16
Diameter of Bolts	–	–	5/8	5/8	3/4	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	1	1 1/8
[2] *250 lb flanges have a 1/16" raised face which is included in the flange thickness dimensions.*															
150 lb BRONZE – ANSI															
Diameter of Flange	3 1/2	3 7/8	4 1/4	4 5/8	5	6	7	7 1/2	8 1/2	9	10	11	13 1/2	16	19
Thickness of Flange (min)[3]	5/16	11/32	3/8	13/32	7/16	1/2	9/16	5/8	11/16	11/16	3/4	13/16	15/16	1	1 1/16
Diameter of Bolt Circle	2 3/8	2 3/4	3 1/8	3 1/2	3 7/8	4 3/4	5 1/2	6	7	7 1/2	8 1/2	9 1/2	11 3/4	14 1/4	17
Number of Bolts	4	4	4	4	4	4	4	4	8	8	8	8	8	12	12
Diameter of Bolts	1/2	1/2	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	7/8	7/8
[3] *150 lb bronze flanges have plain faces with two concentric gasket-retaining grooves between the port and the bolt holes.*															
300 lb BRONZE – ANSI															
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	–	–
Thickness of Flange (min)[4]	1/2	17/32	19/32	5/8	11/16	3/4	13/16	29/32	31/32	1 1/16	1 1/8	1 3/16	1 3/8	–	–
Diameter of Bolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	–	–
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	–	–
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	–	–
[4] *300 lb bronze flanges have plain faces with two concentric gasket-retaining grooves between the port and the bolt holes.*															
150 lb STEEL – ANSI															
Diameter of Flange	3 1/2	3 7/8	4 1/4	4 5/8	5	6	7	7 1/2	8 1/2	9	10	11	13 1/2	16	19
Thickness of Flange (min)[5]	–	–	7/16	1/2	9/16	5/8	11/16	3/4	13/16	15/16	15/16	1	1 1/8	1 3/16	1 1/4
Diameter of Raised Face	1 3/8	1 11/16	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	2 3/8	2 3/4	3 1/8	3 1/2	3 7/8	4 3/4	5 1/2	6	7	7 1/2	8 1/2	9 1/2	11 3/4	14 1/4	17
Number of Bolts	4	4	4	4	4	4	4	4	8	8	8	8	8	12	12
Diameter of Bolts	1/2	1/2	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	7/8	7/8
[5] *150 lb steel flanges have a 1/16"raised face which is included in the flange thickness dimensions.*															
300 lb STEEL – ANSI															
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	17 1/2	20 1/2
Thickness of Flange (min)[6]	–	–	11/16	3/4	13/16	7/8	1	1 1/8	1 3/16	1 1/4	1 3/8	1 7/16	1 5/8	1 7/8	2
Diameter of Raised Face	1 3/8	1 11/16	2	2 1/2	2-7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	15 1/4	17 3/4
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	16	16
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	1	1 1/8
[6] *300 lb steel flanges have a 1/16" raised face which is included in the flange thickness dimensions.*															
400 lb STEEL – ANSI															
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	17 1/2	20 1/2
Thickness of Flange (min)[7]	9/16	5/8	11/16	13/16	7/8	1	1 1/8	1 1/4	1 3/8	1 3/8	1 1/2	1 5/8	1 7/8	2 1/8	2 1/4
Diameter of Raised Face	1 3/8	1 11/16	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	15 1/4	17 3/4
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	16	16
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	7/8	7/8	7/8	7/8	1	1 1/8	1 1/4
[7] *400 lb steel flanges have a 1/4" raised face which is NOT included in the flange thickness dimensions.*															
600 lb STEEL – ANSI															
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10 3/4	13	14	16 1/2	20	22
Thickness of Flange (min)[8]	9/16	5/8	11/16	13/16	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	1 7/8	2 3/16	2 1/2	2 5/8
Diameter of Raised Face	1 3/8	1 11/16	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter ofBolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	8 1/2	10 1/2	11 1/2	13 3/4	17	19 1/4
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	16	20
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	7/8	7/8	1	1	1 1/8	1 1/4	1 1/4
[8] *600 lb steel flanges have a 1/4" raised face which is NOT included in the flange thickness dimensions.*															

Table 38: Schedule 40 Pipe Dimensions

	Diameters			Transverse Areas			Length of Pipe per Sq. Foot of				
Size Inches	External Inches	Internal Inches	Nominal Thickness Inches	External Sq. Ins.	Internal Sq. Ins.	Metal Sq. Ins.	External Surface Feet	Internal Surface Feet	Cubic Feet per Foot of Pipe	Weight per Foot Pounds	Number Threads per Inch of Screw
1/8	.405	.269	.068	.129	.057	.072	9.431	14.199	.00039	.244	27
1/4	.540	.364	.088	.229	.104	.125	7.073	10.493	.00072	.424	18
3/8	.675	.493	.091	.358	.191	.167	5.658	7.747	.00133	.567	18
1/2	.840	.622	.109	.554	.304	.250	4.547	6.141	.00211	.850	14
3/4	1.050	.824	.113	.866	.533	.333	3.637	4.635	.00370	1.130	14
1	1.315	1.049	.133	1.358	.864	.494	2.904	3.641	.00600	1.678	11 1/2
1 1/4	1.660	1.380	.140	2.164	1.495	.669	2.301	2.767	.01039	2.272	11 1/2
1 1/2	1.900	1.610	.145	2.835	2.036	.799	2.010	2.372	.01414	2.717	11 1/2
2	2.375	2.067	.154	4.430	3.355	1.075	1.608	1.847	.02330	3.652	11 1/2
2 1/2	2.875	2.469	.203	6.492	4.788	1.704	1.328	1.547	.03325	5.793	8
3	3.500	3.068	.216	9.621	7.393	2.228	1.091	1.245	.05134	7.575	8
3 1/2	4.000	3.548	.226	12.56	9.886	2.680	.954	1.076	.06866	9.109	8
4	4.500	4.026	.237	15.90	12.73	3.174	.848	.948	.08840	10.790	8
5	5.563	5.047	.258	24.30	20.00	4.300	.686	.756	.1389	14.61	8
6	6.625	6.065	.280	34.47	28.89	5.581	.576	.629	.2006	18.97	8
8	8.625	7.981	.322	58.42	50.02	8.399	.442	.478	.3552	28.55	8
10	10.750	10.020	.365	90.76	78.85	11.90	.355	.381	.5476	40.48	8
12	12.750	11.938	.406	127.64	111.9	15.74	.299	.318	.7763	53.6	
14	14.000	13.125	.437	153.94	135.3	18.64	.272	.280	.9354	63.0	
16	16.000	15.000	.500	201.05	176.7	24.35	.238	.254	1.223	78.0	
18	18.000	16.874	.563	254.85	224.0	30.85	.212	.226	1.555	105.0	
20	20.000	18.814	.593	314.15	278.0	36.15	.191	.203	1.926	123.0	
24	24.000	22.626	.687	452.40	402.1	50.30	.159	.169	2.793	171.0	

Table 39: Schedule 80 Pipe Dimensions

	Diameters			Transverse Areas			Length of Pipe per Sq. Foot of				
Size Inches	External Inches	Internal Inches	Nominal Thickness Inches	External Sq. Ins.	Internal Sq. Ins.	Metal Sq. Ins.	External Surface Feet	Internal Surface Feet	Cubic Feet per Foot of Pipe	Weight per Foot Pounds	Number Threads per Inch of Screw
1/8	.405	.215	.095	.129	.036	.093	9.431	17.750	.00025	.314	27
1/4	.540	.302	.119	.229	.072	.157	7.073	12.650	.00050	.535	18
3/8	.675	.423	.126	.358	.141	.217	5.658	9.030	.00098	.738	18
1/2	.840	.546	.147	.554	.234	.320	4.547	7.000	.00163	1.00	14
3/4	1.050	.742	1.54	.866	.433	.433	3.637	5.15	.00300	1.47	14
1	1.315	.957	.179	1.358	.719	.639	2.904	3.995	.00500	2.17	11 1/2
1 1/4	1.660	1.278	.191	2.164	1.283	.881	2.301	2.990	.00891	3.00	11 1/2
1 1/2	1.900	1.500	.200	2.835	1.767	1.068	2.010	2.542	.01227	3.65	11 1/2
2	2.375	1.939	.218	4.430	2.953	1.477	1.608	1.970	.02051	5.02	11 1/2
2 1/2	2.875	2.323	.276	6.492	4.238	2.254	1.328	1.645	.02943	7.66	8
3	3.500	2.900	.300	9.621	6.605	3.016	1.091	1.317	.04587	10.3	8
3 1/2	4.000	3.364	.318	12.56	8.888	3.678	.954	1.135	.06172	12.5	8
4	4.500	3.826	.337	15.90	11.497	4.407	.848	.995	.0798	14.9	8
5	5.563	4.813	.375	24.30	18.194	6.112	.686	.792	.1263	20.8	8
6	6.625	5.761	.432	34.47	26.067	8.300	.576	.673	.1810	28.6	8
8	8.625	7.625	.500	58.42	45.663	12.76	.442	.501	.3171	43.4	8
10	10.750	9.564	.593	90.76	71.84	18.92	.355	.400	.4989	64.4	8
12	12.750	11.376	.687	127.64	101.64	26.00	.299	.336	.7058	88.6	
14	14.000	12.500	.750	153.94	122.72	31.22	.272	.306	.8522	107.0	
16	16.000	14.314	.843	201.05	160.92	40.13	.238	.263	1.117	137.0	
18	18.000	16.126	.937	254.85	204.24	50.61	.212	.237	1.418	171.0	
20	20.000	17.938	1.031	314.15	252.72	61.43	.191	.208	1.755	209.0	
24	24.000	21.564	1.218	452.40	365.22	87.18	.159	.177	2.536	297.0	

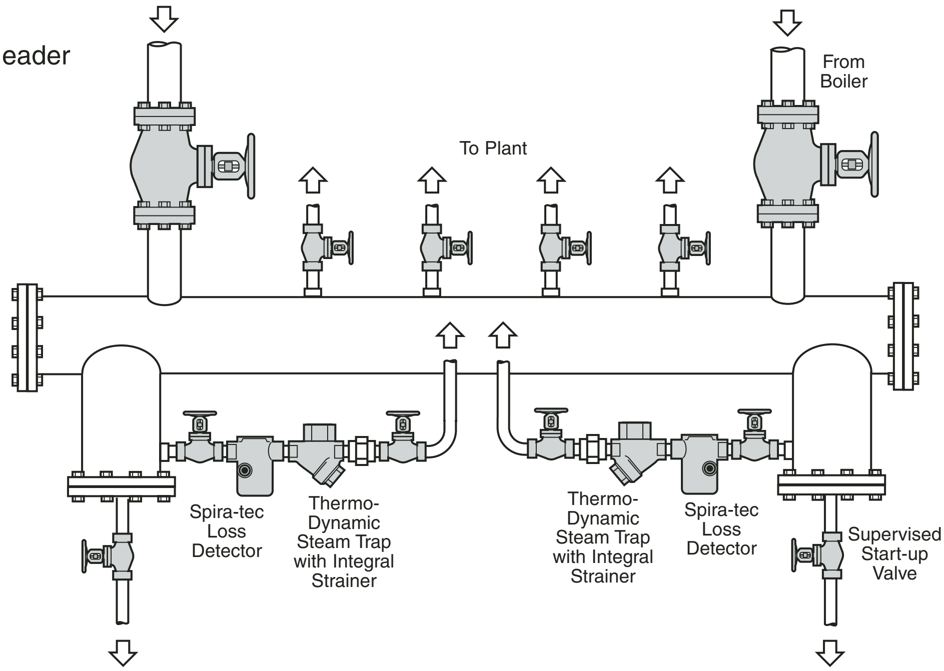
HOOK-UP APPLICATION DIAGRAMS

Section 2

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Figure II-1

Boiler Steam Header



Boiler Steam Headers provide collecting vessels for the steam flowing from one or more boilers, and distribute it to as many mains as are needed to supply the plant. Often the flow may be in either direction along the header depending on which boilers and which supply lines are being used. Selecting the ideal location for the drip point is thus complicated. It is recommended to make the header of such an increased diameter as to drop the steam velocity through it to a low value even with maximum flow in either direction. The header can then act also as a separator, and generously sized steam traps can be fitted at each end.

The boiler header and the separator, which should be fitted in the steam take off from modern high performance packaged boilers, may sometimes have to cope with carry-over from the boiler. These two locations form the exception to the general rule that mains drip points rarely need a steam trap as large as the 1/2" size and can usually be fitted with 1/2" Low Capacity traps. Instead, traps in 3/4" and even 1" sizes are often used. The potential for steam losses when these larger traps eventually become worn is increased, and the use of Spira-tec steam trap monitors is especially valid.

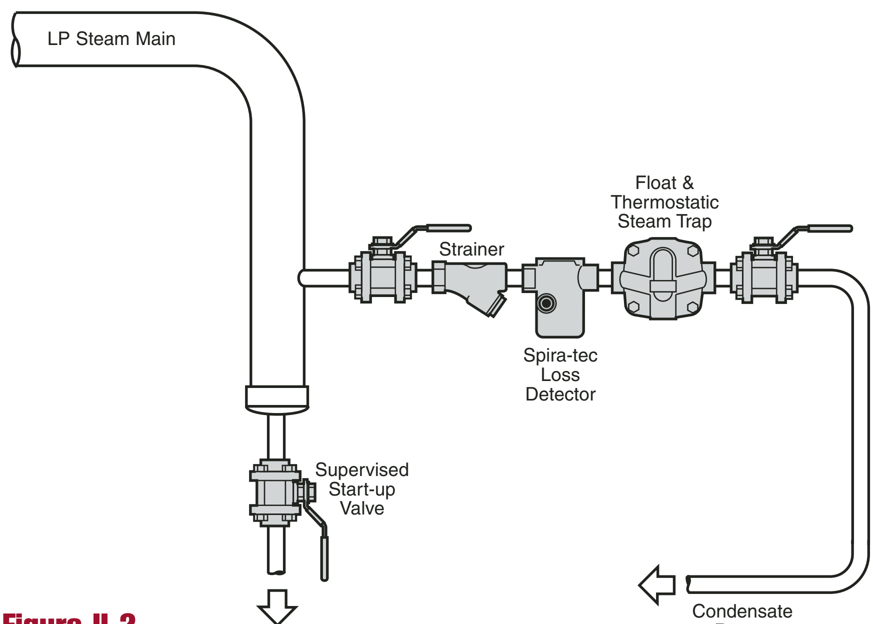


Figure II-2

Draining End of Low Pressure Steam Main

In the case of low pressure mains, the use of Float and Thermostatic traps is recommended for the drip stations. The introduction of F & T traps with steel bodies, third generation capsule type or bimetallic air vents, and operating mechanisms suitable for pressures up to 465 psi, means that F & T traps can also be used on properly drained lines where waterhammer does not occur, even at pressures which would formerly have excluded them. An auxiliary air vent is recommended for the end of all mains where the system is started up automatically.

Figure II-3

Draining and Air Venting Steam Lines

Drip points along the run of the steam lines, and at the bottom of any risers, should incorporate large diameter collecting pockets. Equal tees are useful in sizes up through 6" and larger size pipes can have pockets 2 or 3 sizes smaller than the main but not less than 6". The terminal points of the mains should have automatic air vents, and equal tees again provide convenient collecting pockets for both condensate and air when installed as shown.

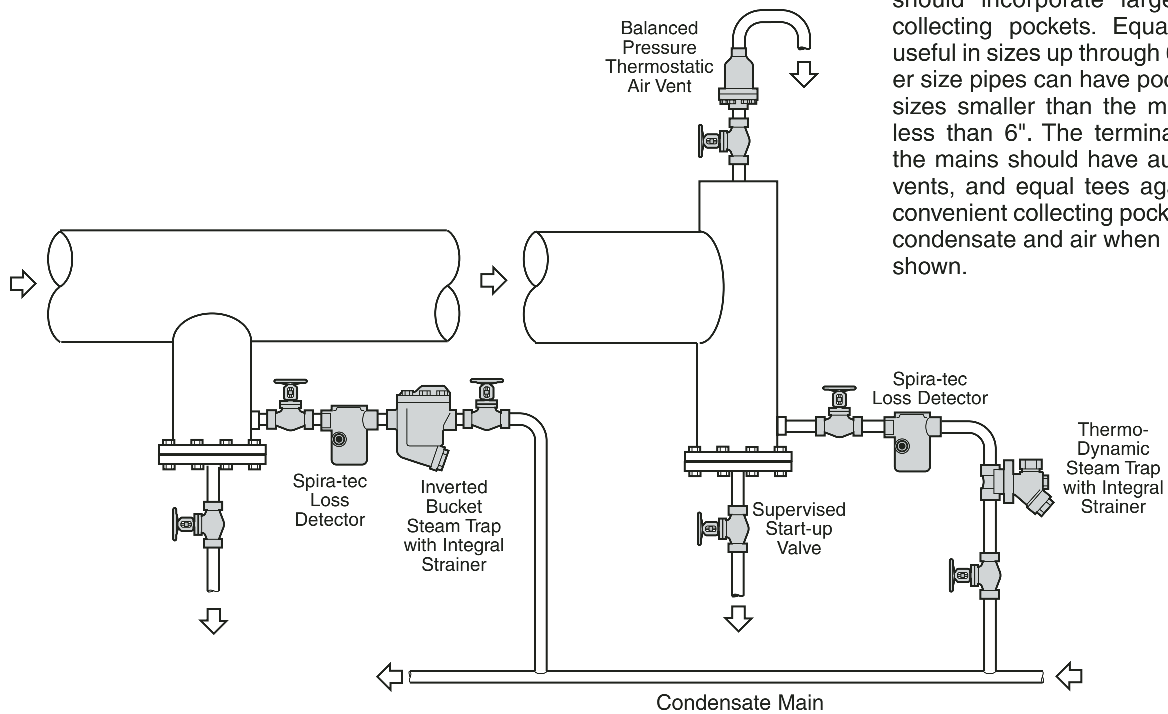
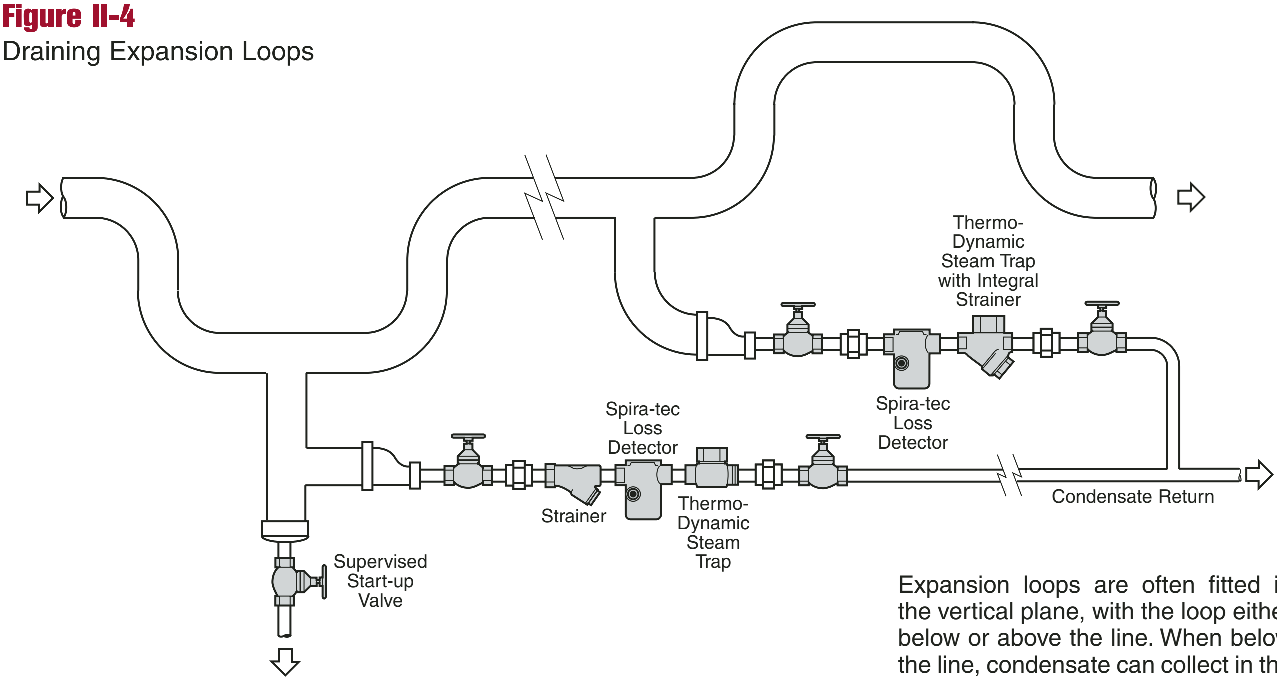


Figure II-4

Draining Expansion Loops



Expansion loops are often fitted in the vertical plane, with the loop either below or above the line. When below the line, condensate can collect in the bottom of the loop. Above the line, it will collect just in front of the loop, at the foot of the riser. Drainage points are necessary in each case, as shown.

Figure II-5

Draining Steam Mains to Return Main at Same Level

Both HP and LP mains often must be drained to a condensate return line at the same elevation as the steam line. The best location for the traps is then below the steam line, with a riser after the trap to the top of the return line.

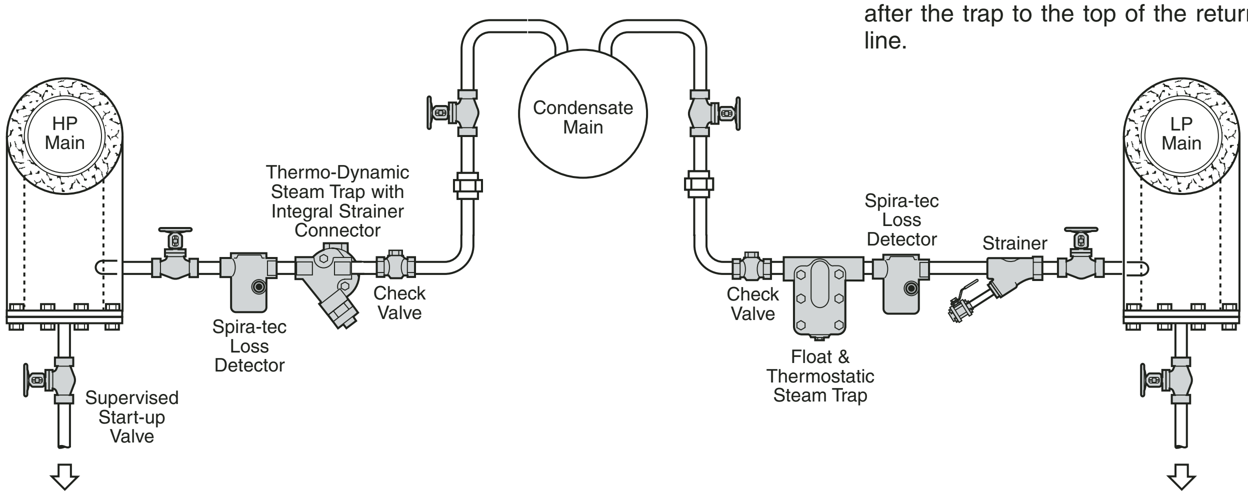


Figure II-6

Trapping Hook-up for Start-up of Steam Main

For supervised startup of steam mains, a manual bypass is fitted so that condensate can be drained by gravity while the line pressure is too low for it to be handled by the trap at an adequate rate. If a second trap is fitted in the bypass line, a similar hookup is obtained which is suitable for automatic startup.

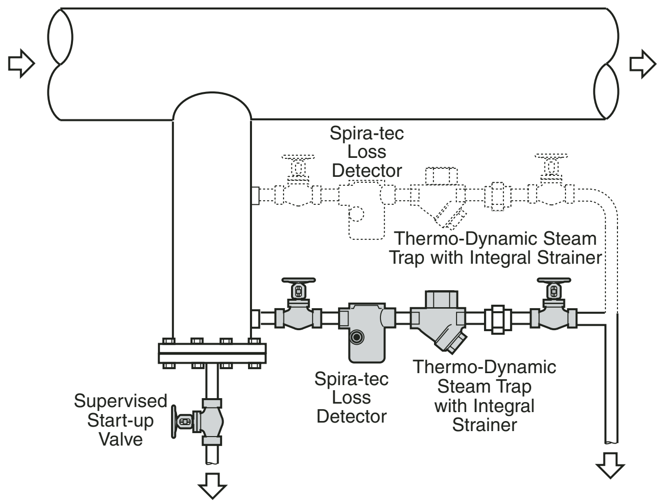


Figure II-7

Hook-up with Condensate Return Line at High Level

Often the normal trap discharges to a return line at higher elevation. The startup trap must always discharge by gravity so here it is separated from the "normal running" trap. A Thermoton is used so that it will close automatically when the condensate temperature shows that warm up of the main is nearing completion.

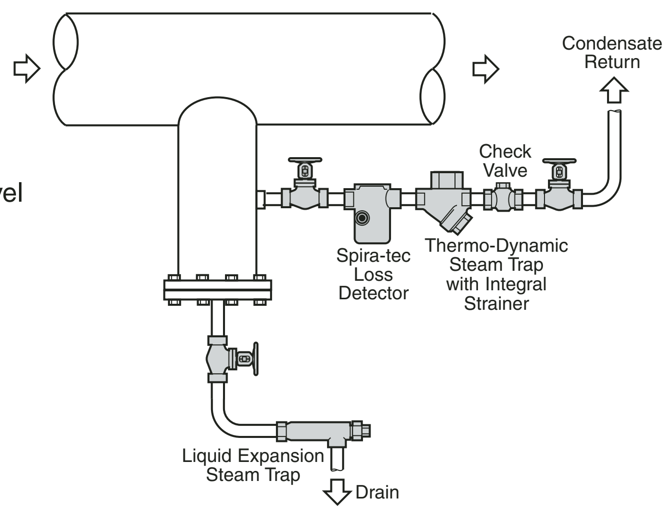
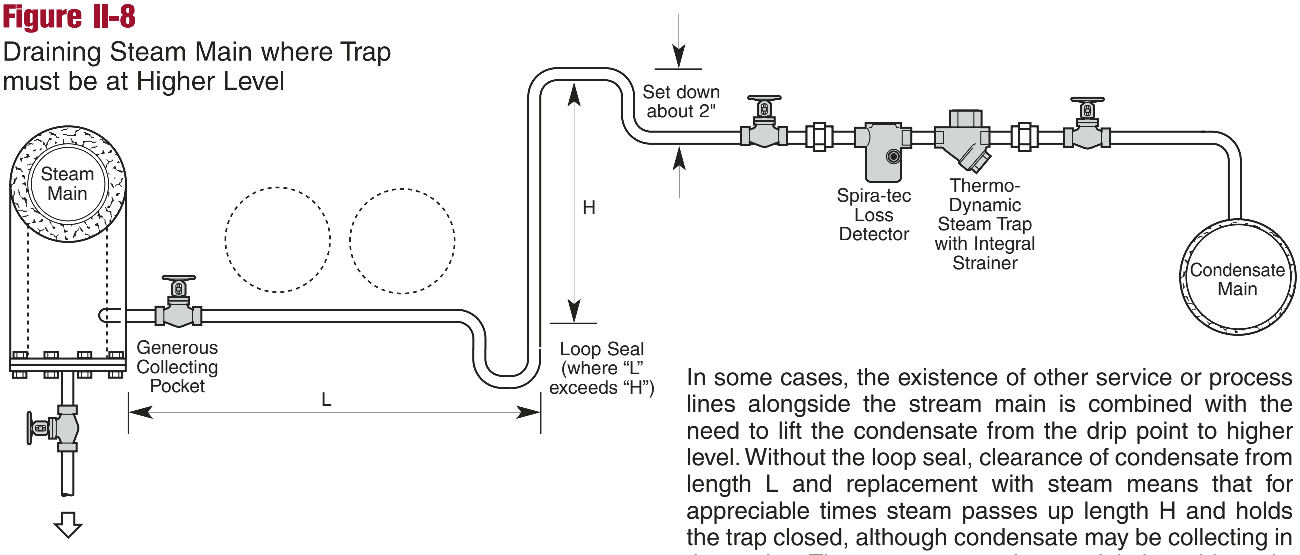


Figure II-8

Draining Steam Main where Trap must be at Higher Level

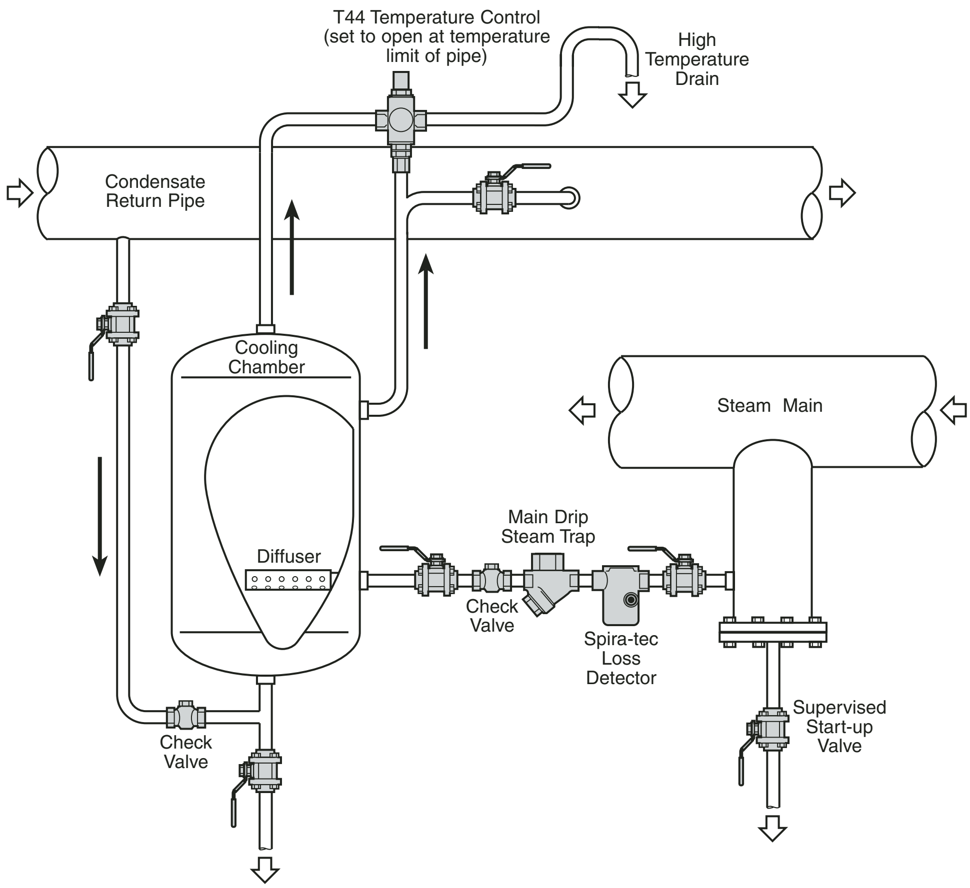


In some cases, the existence of other service or process lines alongside the stream main is combined with the need to lift the condensate from the drip point to higher level. Without the loop seal, clearance of condensate from length L and replacement with steam means that for appreciable times steam passes up length H and holds the trap closed, although condensate may be collecting in the pocket. The arrangement shown minimizes this problem and gives most consistent performance of the trap.

Figure II-9

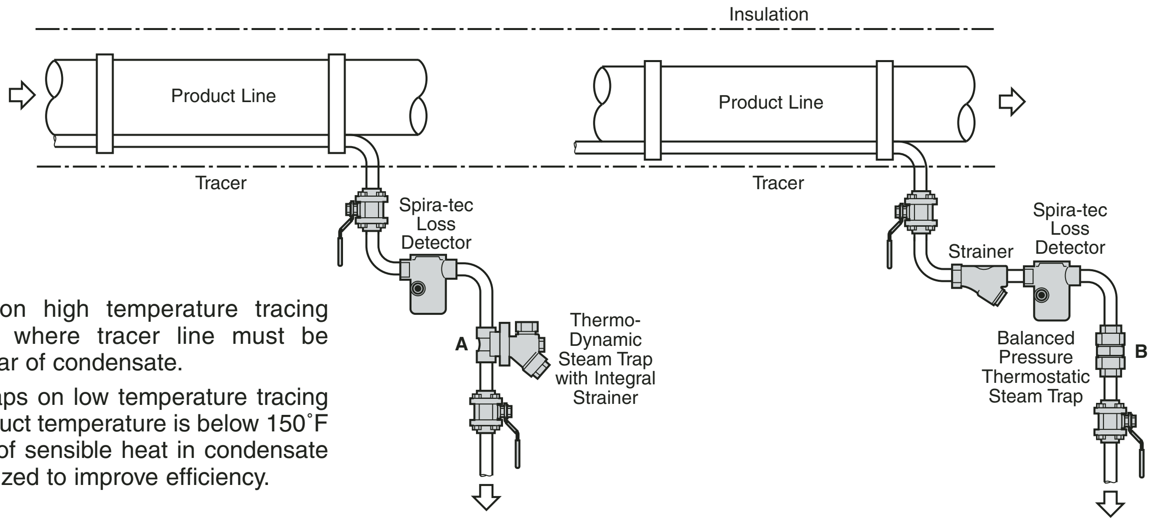
Condensate Drainage to Reinforced Plastic Return Line, with Overheat Protection

On some extended sites, steam distribution is underground and drip points are inside "steam pits". Steam main drip traps should discharge into gravity return systems, but at times it may be necessary to connect them directly to a pumped condensate line. To avoid failure of plastic or fiberglass piping caused by high temperature from steam leakage when eventually the trap becomes worn, a cooling chamber and control as shown can be used. If the temperature of the condensate leaving the chamber ever reaches the safe limit, the control valve opens. Condensate is discharged above grade, where it can be seen, until its temperature falls again below the limiting value.



HOOK-UP DIAGRAMS

Figure II-10 Typical Steam Tracer Trapping Arrangements



A. TD traps on high temperature tracing application where tracer line must be drained clear of condensate.

B. TSS300 traps on low temperature tracing where product temperature is below 150˚F and some of sensible heat in condensate may be utilized to improve efficiency.

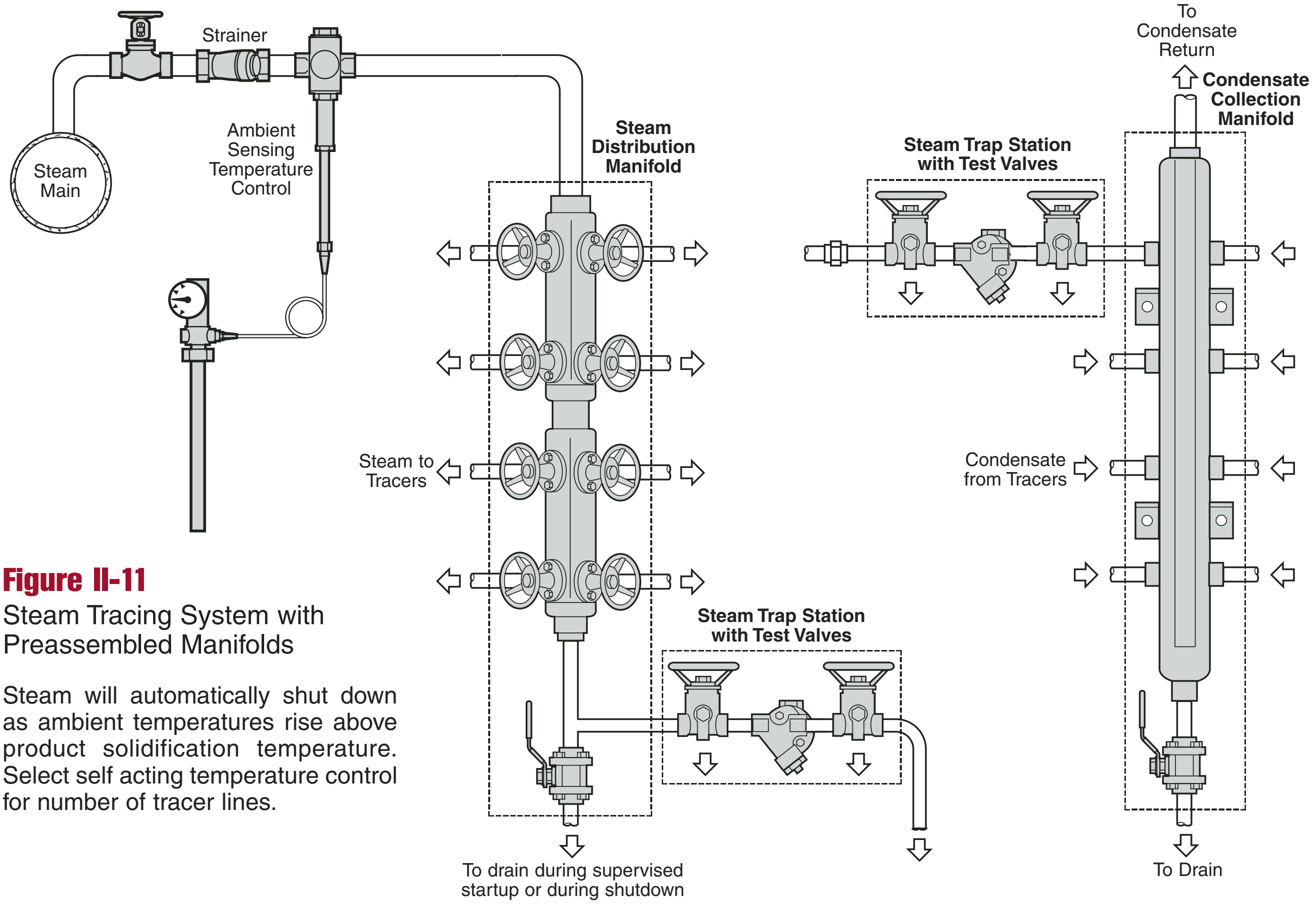


Figure II-11

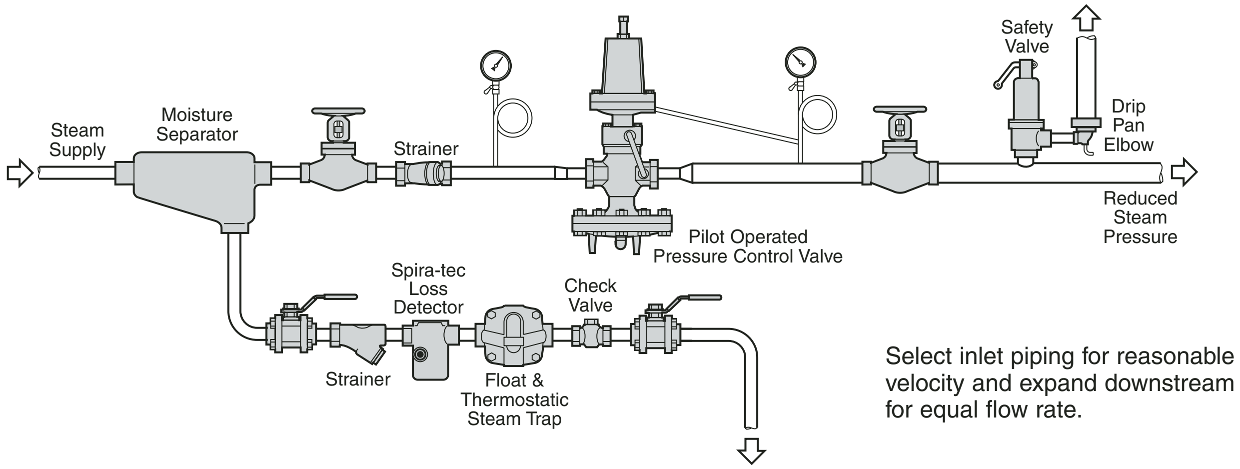
Steam Tracing System with Preassembled Manifolds

Steam will automatically shut down as ambient temperatures rise above product solidification temperature. Select self acting temperature control for number of tracer lines.

Steam is uniformly distributed to tracers by forged steel manifold with integral piston valves. After supplying heat to tracer lines, condensate is collected in fabricated manifold preassembled with steam trap stations. Three-way test valves allow for startup purging, checking of lines for blockage, isolation of trap for maintenance, and visual testing of steam trap operation. Condensate manifold has an internal siphon pipe to reduce waterhammer and provide freeze protection.

Figure II-12

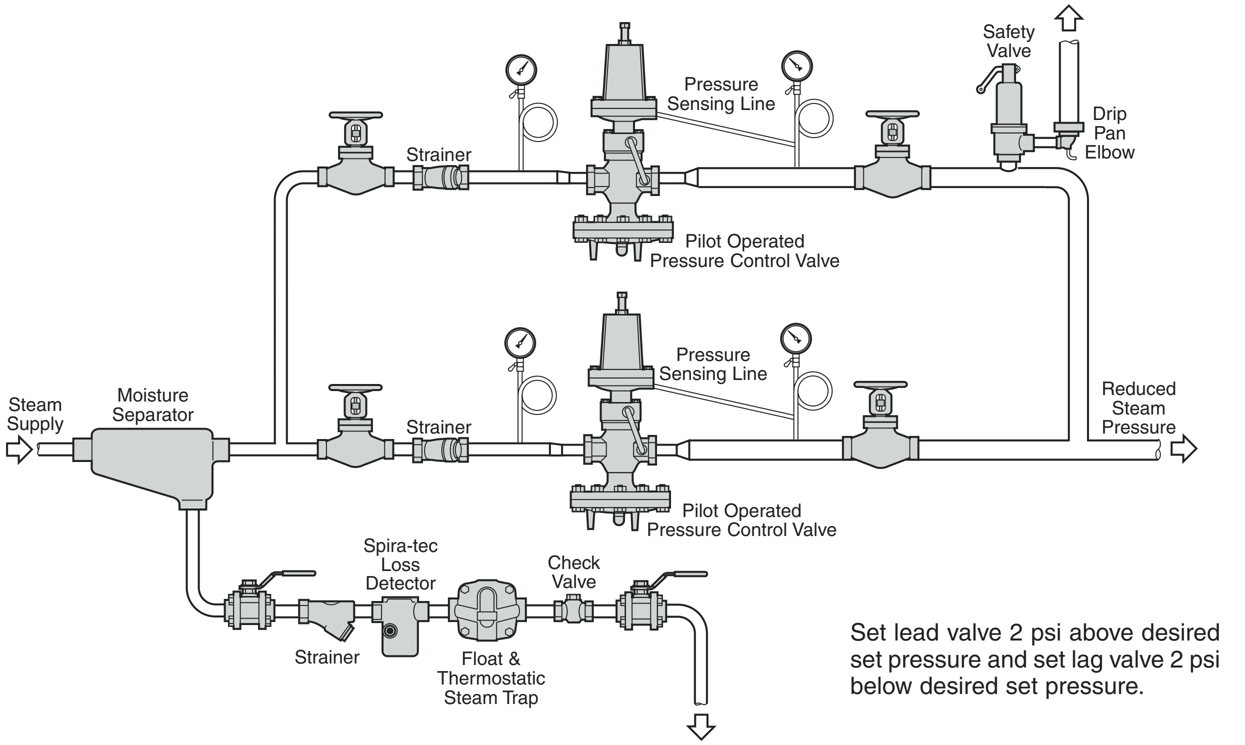
Typical Pressure Reducing Valve Station



Select inlet piping for reasonable velocity and expand downstream for equal flow rate.

Figure II-13

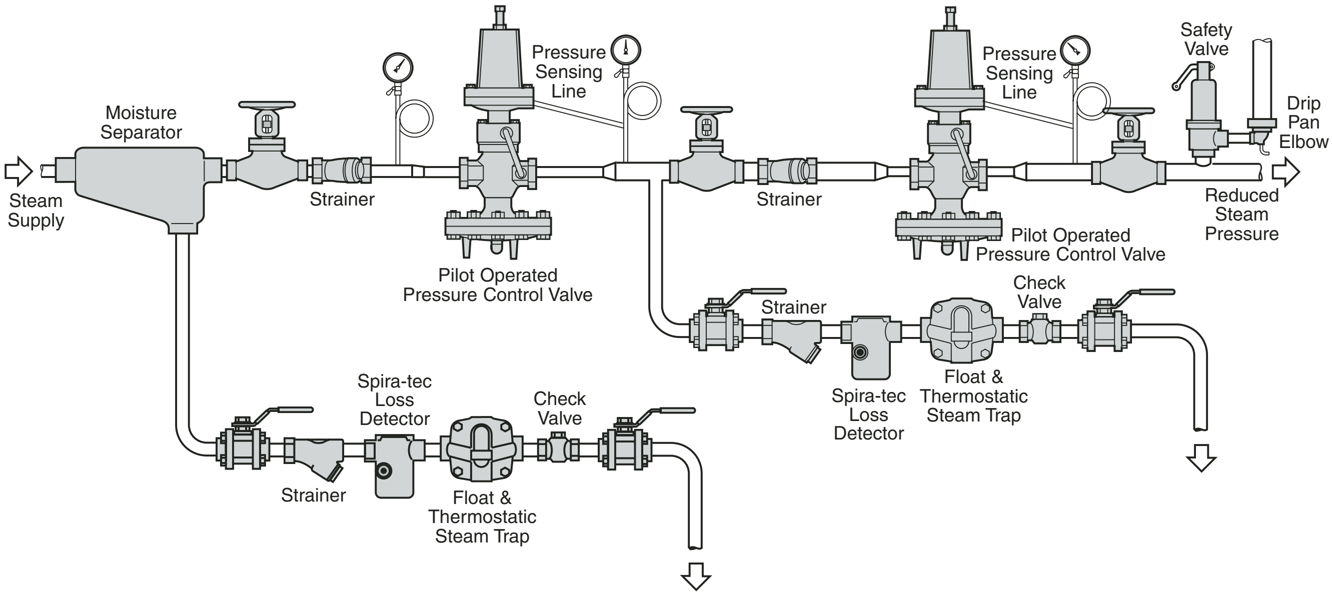
Parallel Operation of Pressure Reducing Valves



Set lead valve 2 psi above desired set pressure and set lag valve 2 psi below desired set pressure.

Figure II-14

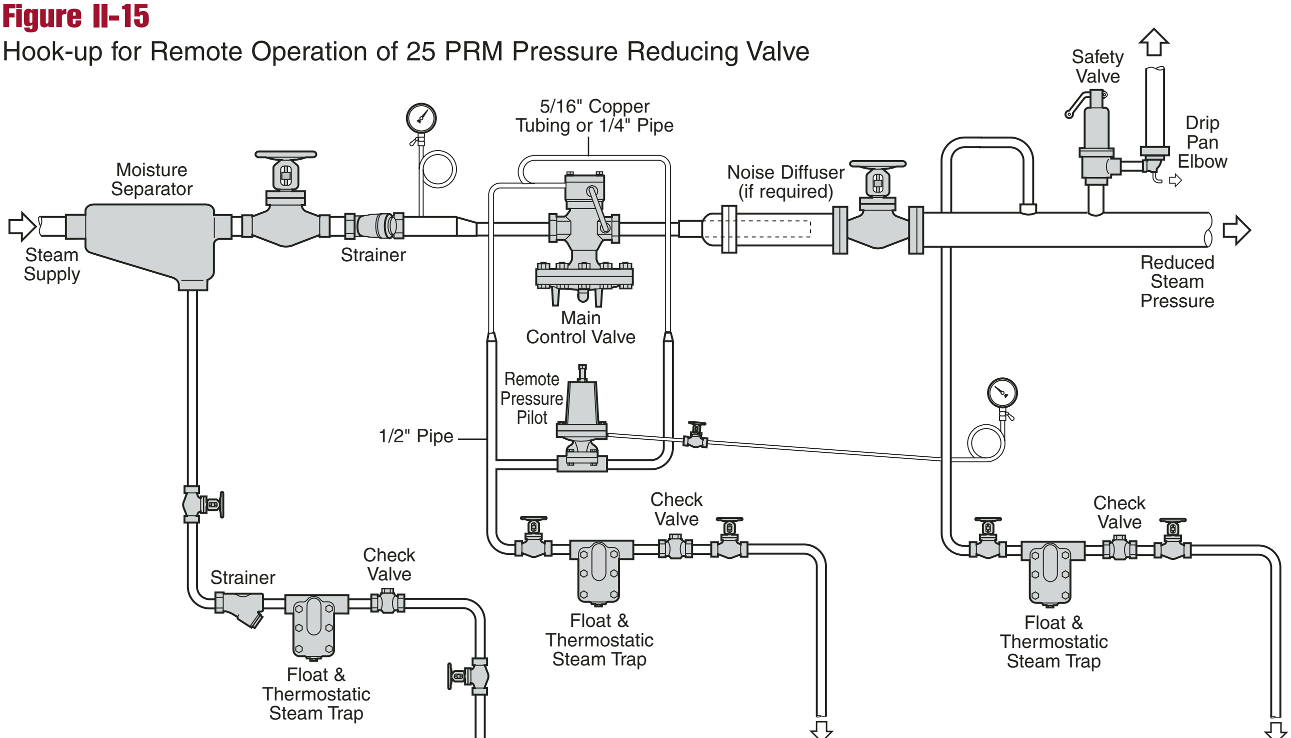
Series Pressure Reducing Valve Station for High Turndown Rations



Note: Intermediate pressure takeoff requires an additional safety valve.

Figure II-15

Hook-up for Remote Operation of 25 PRM Pressure Reducing Valve



Limit pilot to 15 ft. drop below main valve and drain all supply tubing. If pilot is mounted above main valve, pilot line drip traps can be eliminated. For longer distance an air loaded pilot should be used.

Figure II-16

Installation of Pressure Reducing Valve in "Tight Spaces"

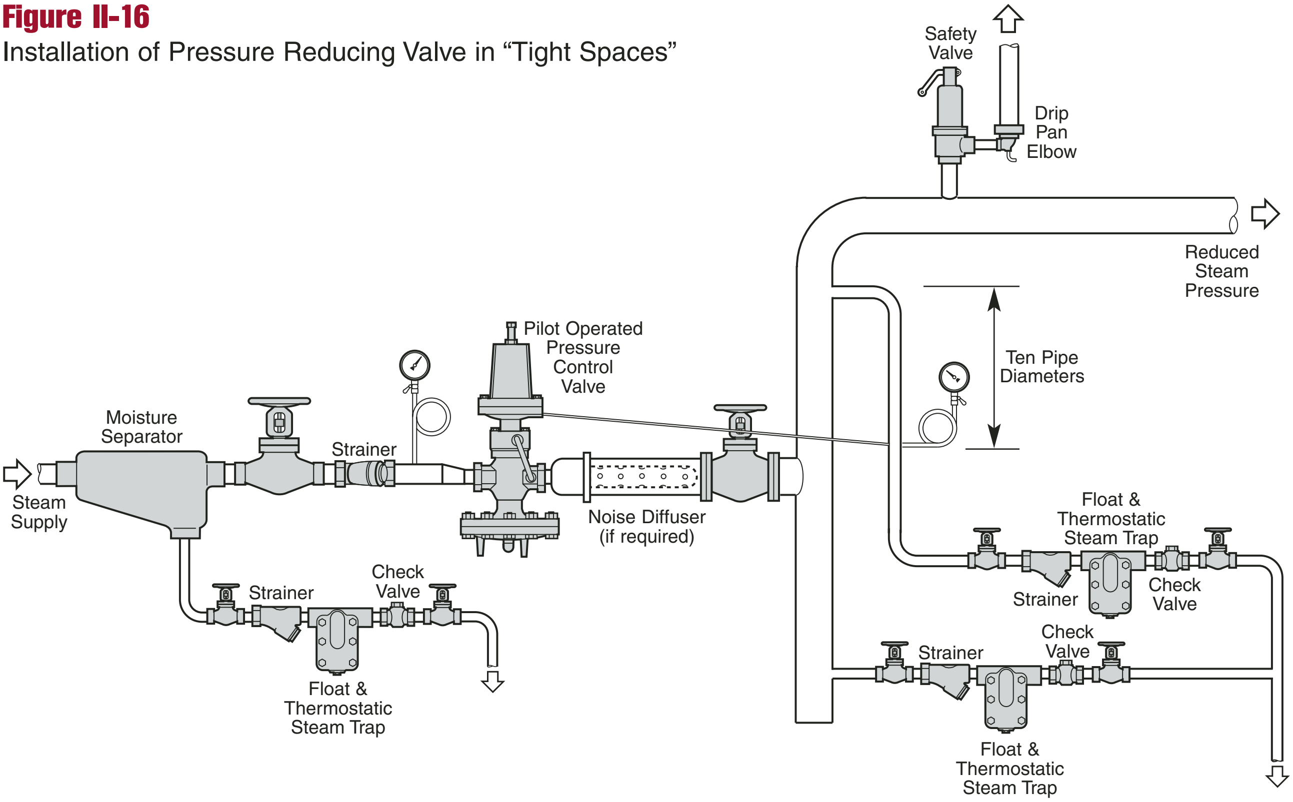


Figure II-17

Low Capacity Pressure Reducing Station

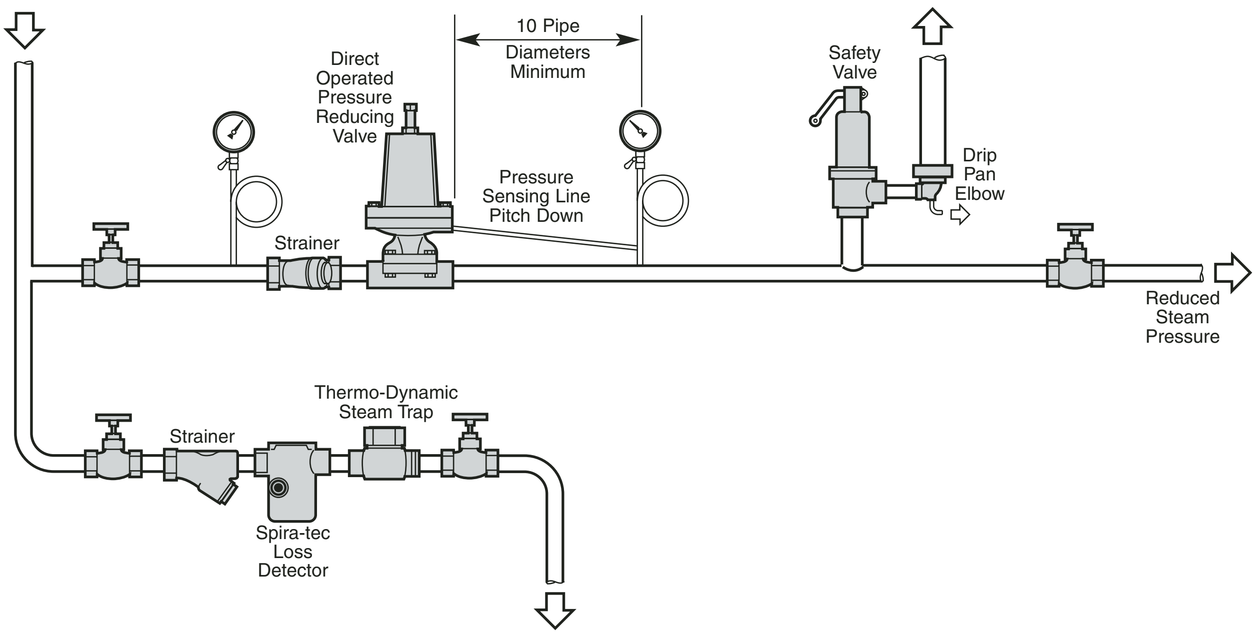
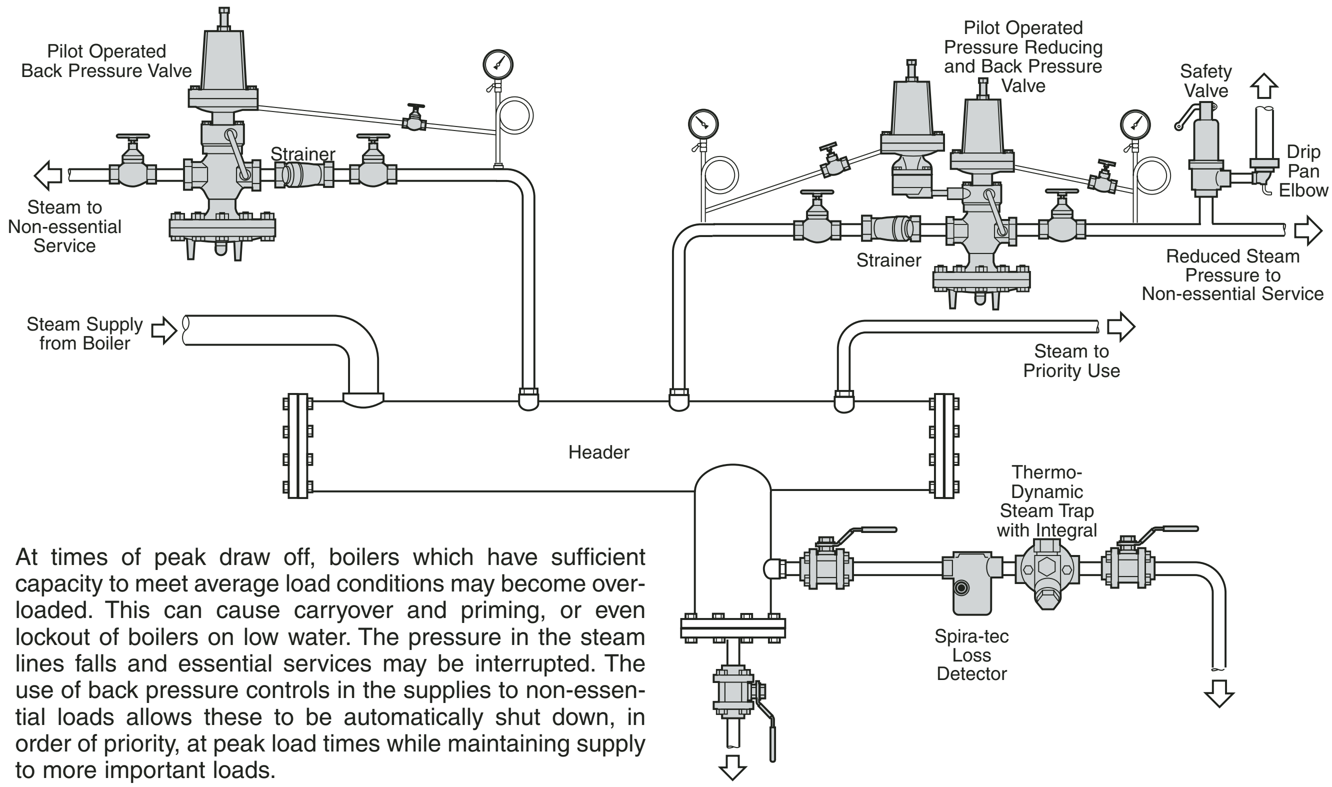


Figure II-18

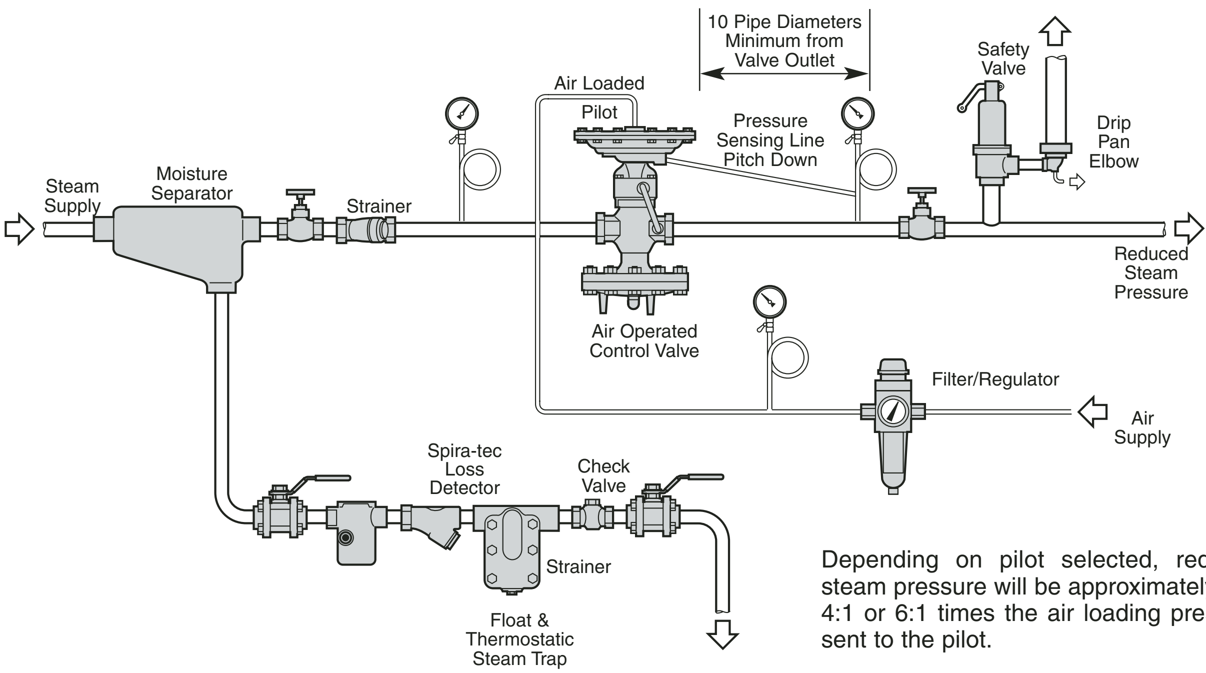
25 BP Back Pressure Controls used to Restrict Supply to Low Priority Uses at Times of Overload



At times of peak draw off, boilers which have sufficient capacity to meet average load conditions may become overloaded. This can cause carryover and priming, or even lockout of boilers on low water. The pressure in the steam lines falls and essential services may be interrupted. The use of back pressure controls in the supplies to non-essential loads allows these to be automatically shut down, in order of priority, at peak load times while maintaining supply to more important loads.

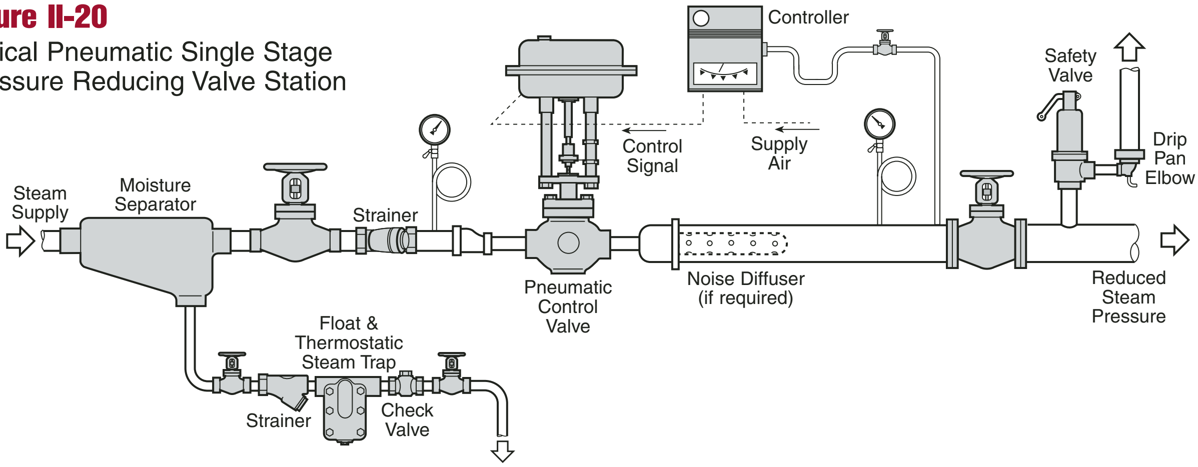
Figure II-19

Reducing Steam Pressure Using 25PA Control Valve with Remote Air Valve



Depending on pilot selected, reduced steam pressure will be approximately 1:1, 4:1 or 6:1 times the air loading pressure sent to the pilot.

Figure II-20
Typical Pneumatic Single Stage Pressure Reducing Valve Station



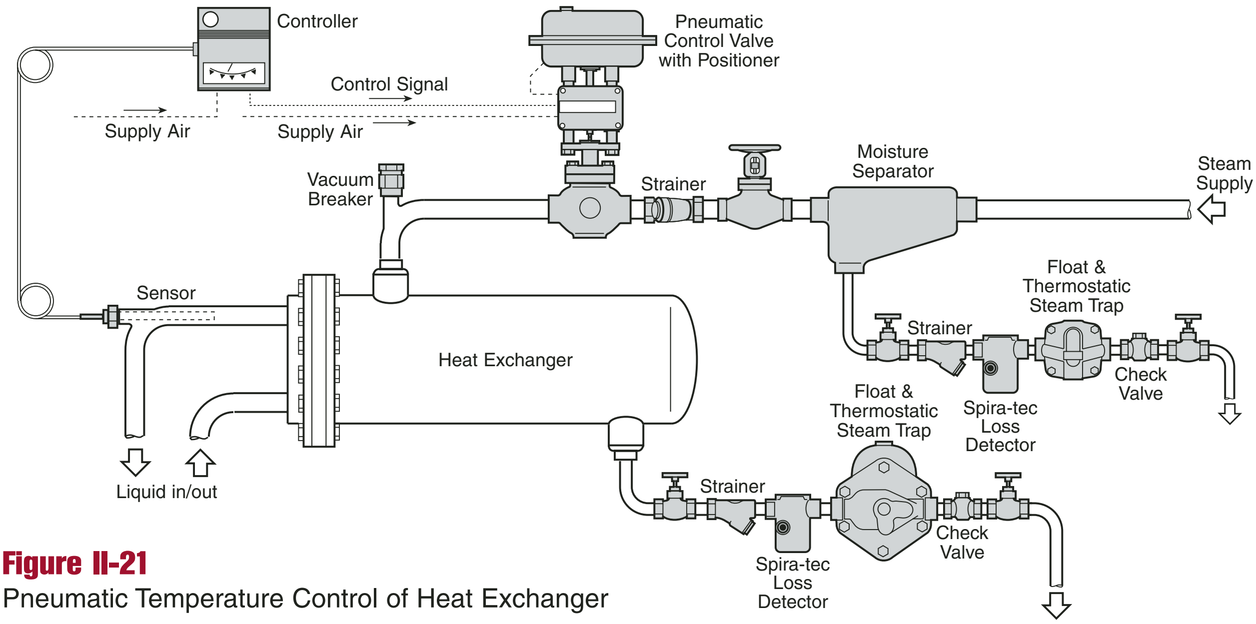


Figure II-21
Pneumatic Temperature Control of Heat Exchanger

Figure II-22
Hook-up for 25 TRM Temperature Control Remotely Mounted (within 15 ft. of Main Valve)

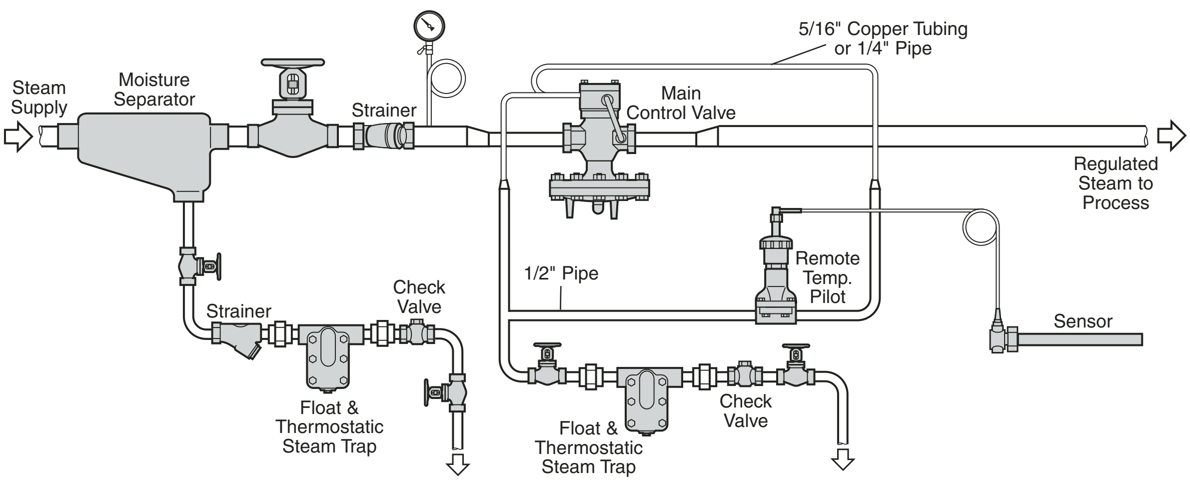


Figure II-23

Pressure Reducing Valve for Pressure Powered Pump Motive Steam

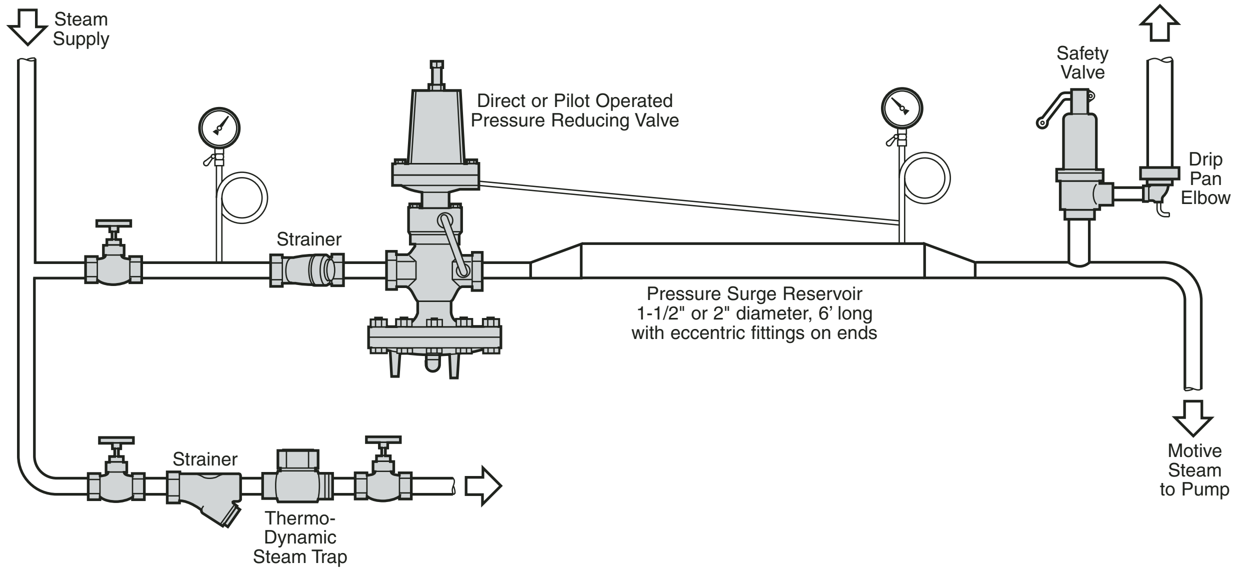
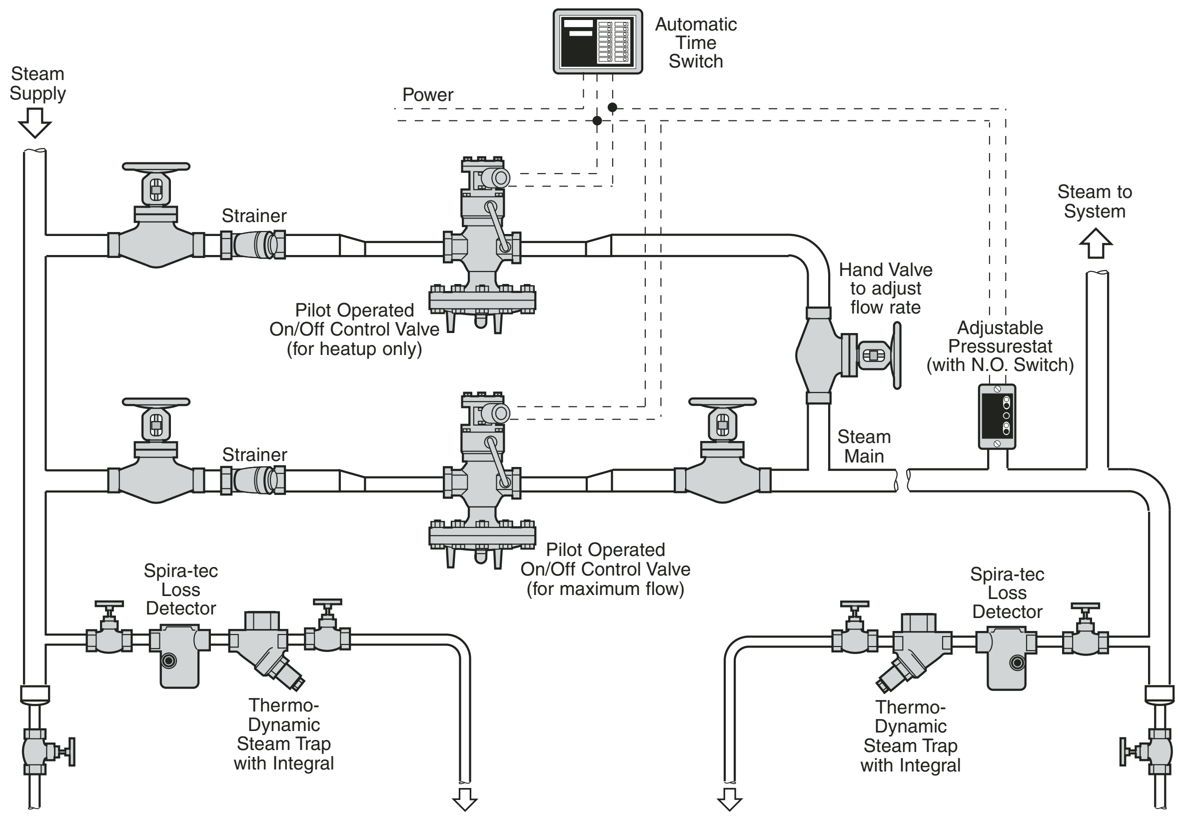


Figure II-24

Heat-up, Pressuring and Shutdown of Steam Mains using On/Off Control Valves and Programmer



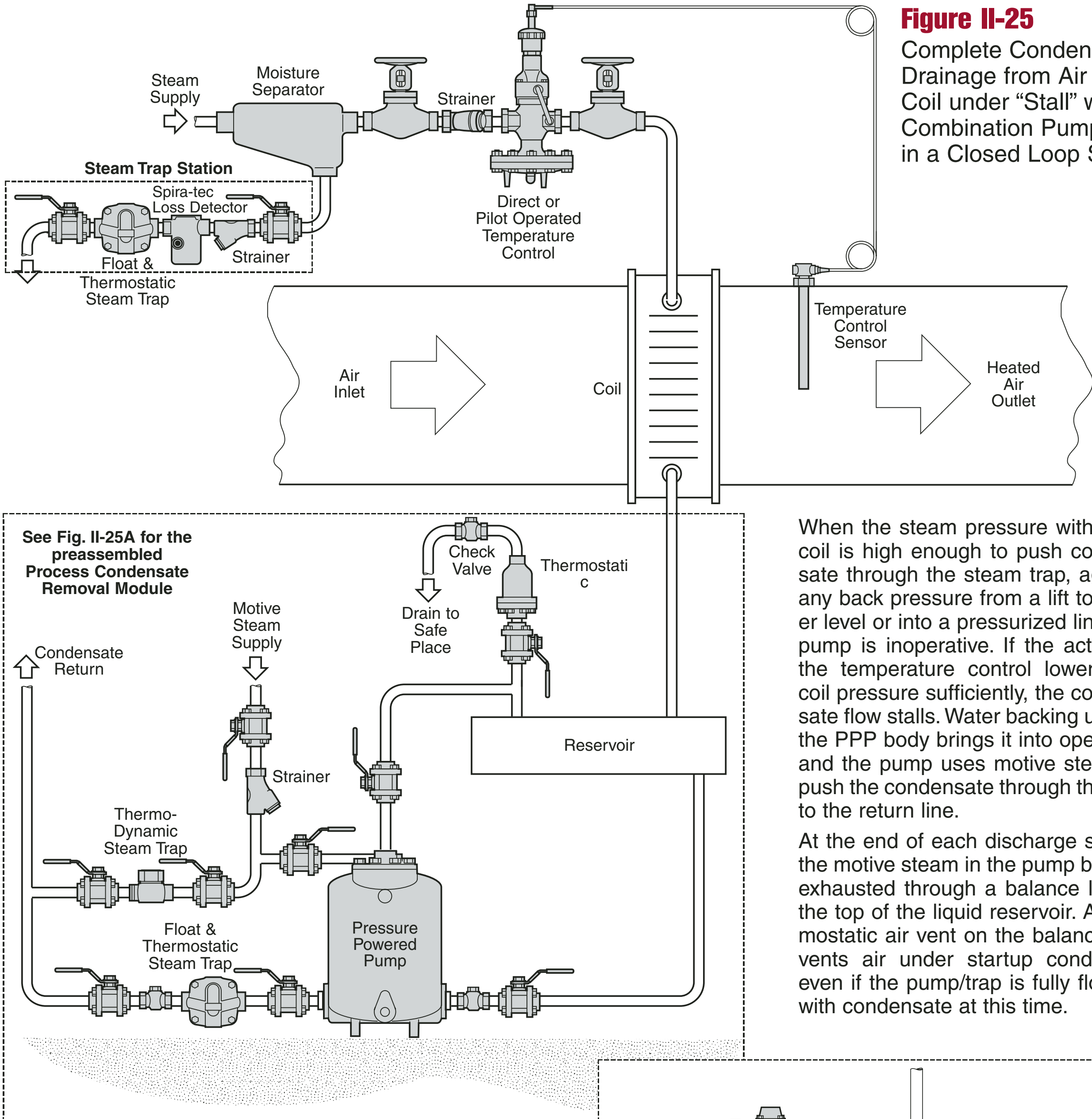


Figure II-25

Complete Condensate Drainage from Air Heater Coil under "Stall" with Combination Pump/Trap in a Closed Loop System

When the steam pressure within the coil is high enough to push condensate through the steam trap, against any back pressure from a lift to higher level or into a pressurized line, the pump is inoperative. If the action of the temperature control lowers the coil pressure sufficiently, the condensate flow stalls. Water backing up into the PPP body brings it into operation and the pump uses motive steam to push the condensate through the trap to the return line.

At the end of each discharge stroke, the motive steam in the pump body is exhausted through a balance line to the top of the liquid reservoir. A thermostatic air vent on the balance line vents air under startup conditions, even if the pump/trap is fully flooded with condensate at this time.

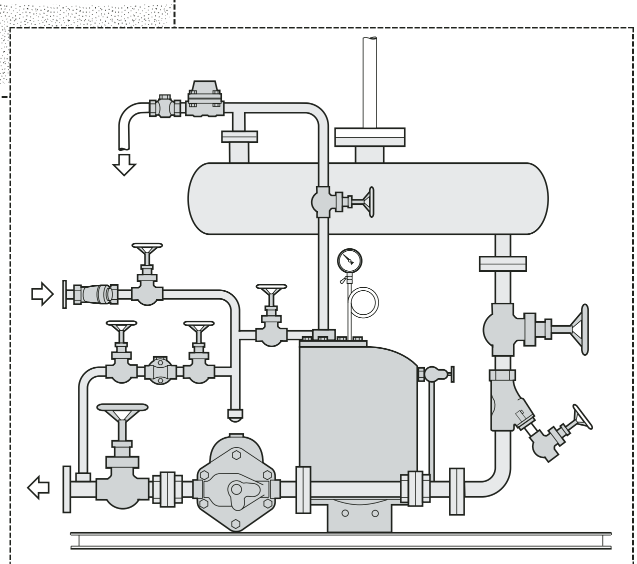


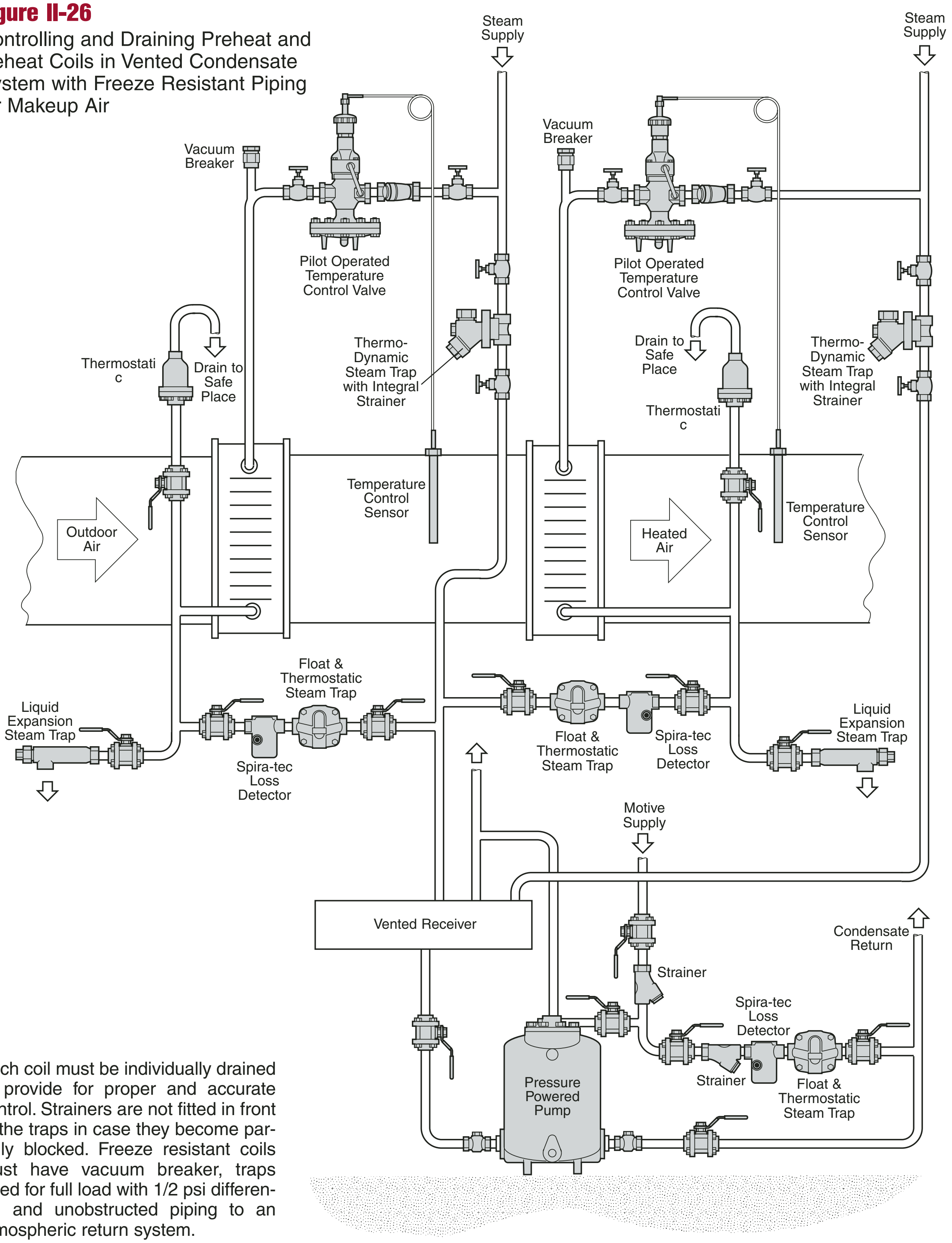
Figure II-25A

Process Condensate Removal Module

A preassembled modular pumping system provides a sole source solution for air heater coil applications.

Figure II-26

Controlling and Draining Preheat and Reheat Coils in Vented Condensate System with Freeze Resistant Piping for Makeup Air



Each coil must be individually drained to provide for proper and accurate control. Strainers are not fitted in front of the traps in case they become partially blocked. Freeze resistant coils must have vacuum breaker, traps sized for full load with 1/2 psi differential and unobstructed piping to an atmospheric return system.

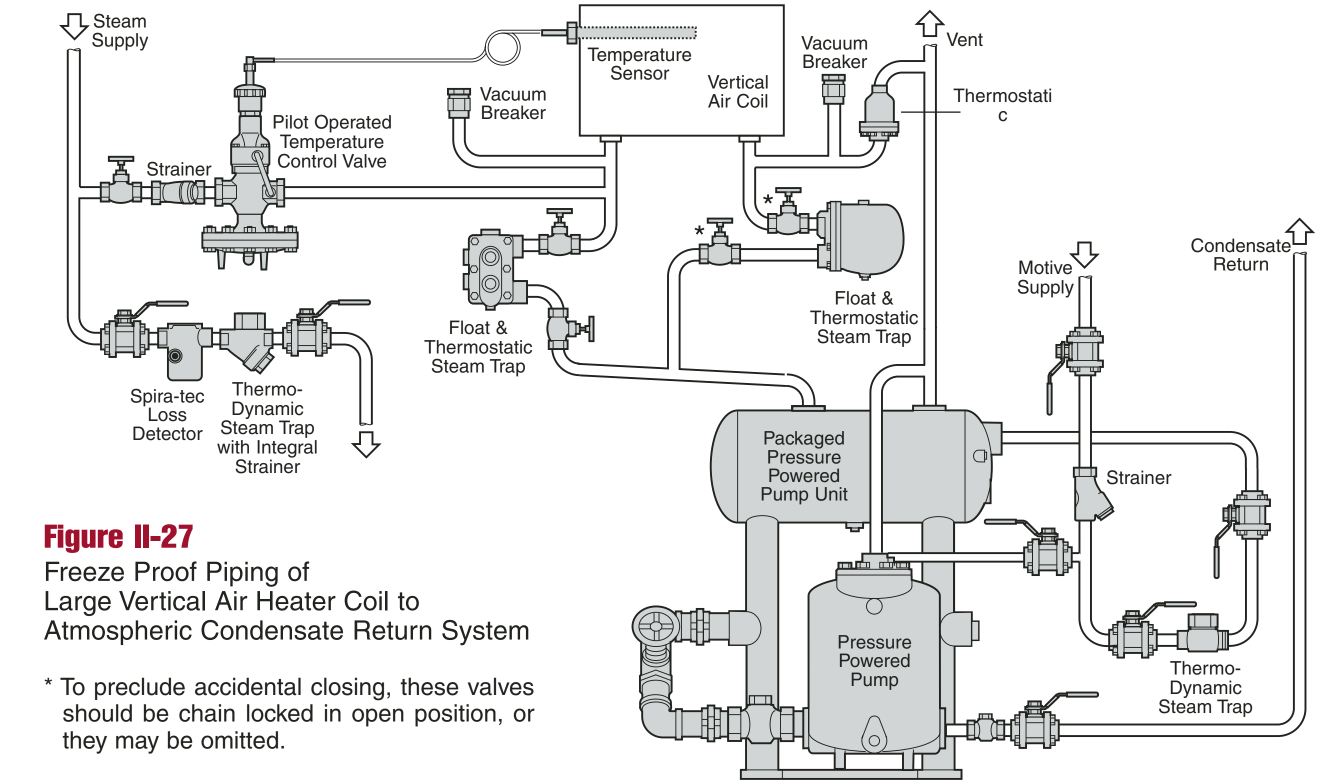


Figure II-27

Freeze Proof Piping of Large Vertical Air Heater Coil to Atmospheric Condensate Return System

* To preclude accidental closing, these valves should be chain locked in open position, or they may be omitted.

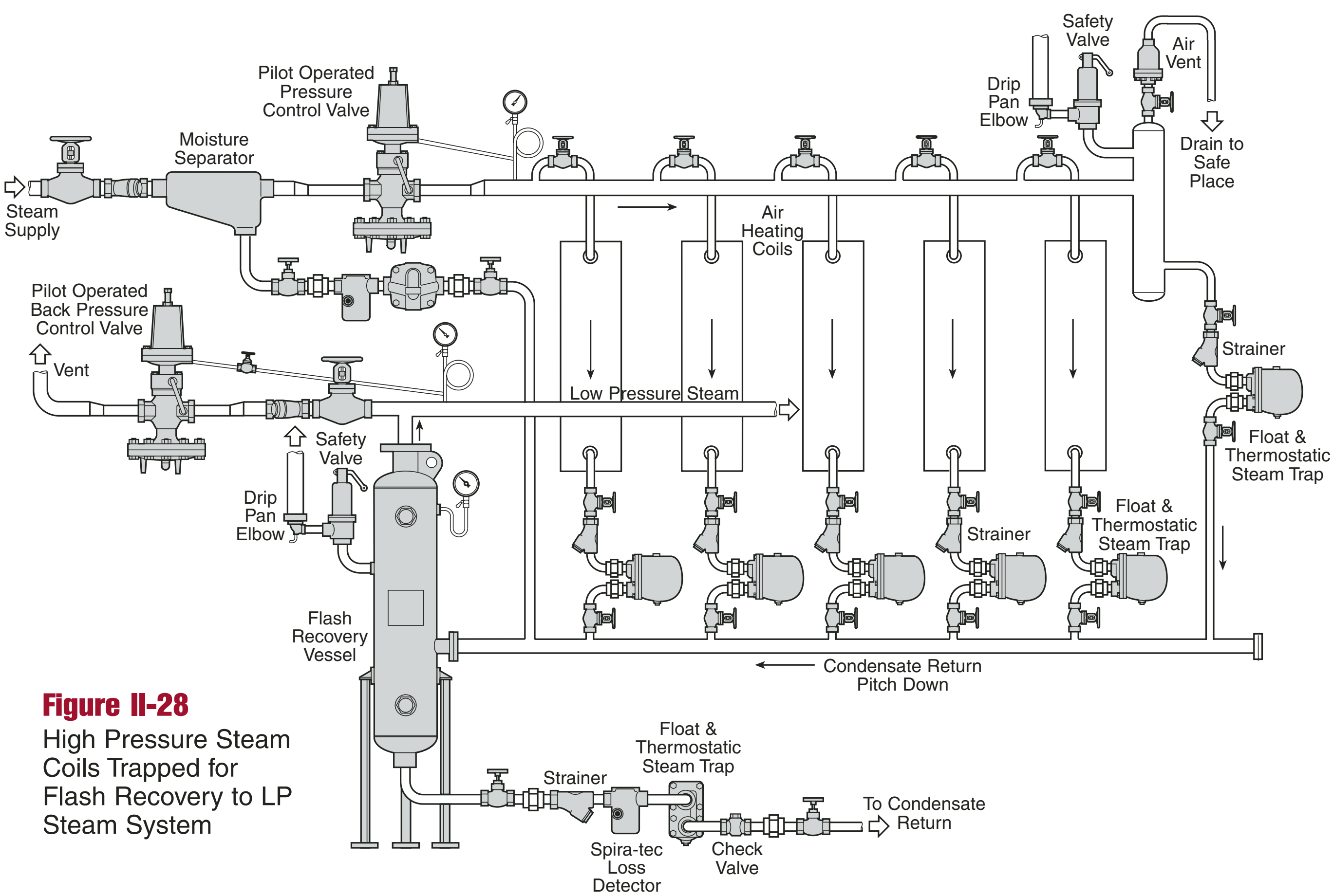


Figure II-28

High Pressure Steam Coils Trapped for Flash Recovery to LP Steam System

Figure II-29
Storage Cylinder with High Limit Protection

Fail safe protection against excess temperatures is provided by a separate control valve, normally latched wide open. If the 130 self-acting control system detects a temperature overrun, or if the control system itself is damaged, a powerful spring is released in the HL10 unit and the high limit valve is driven closed. A switch is available as an extra to provide electrical warning that the device has been actuated.

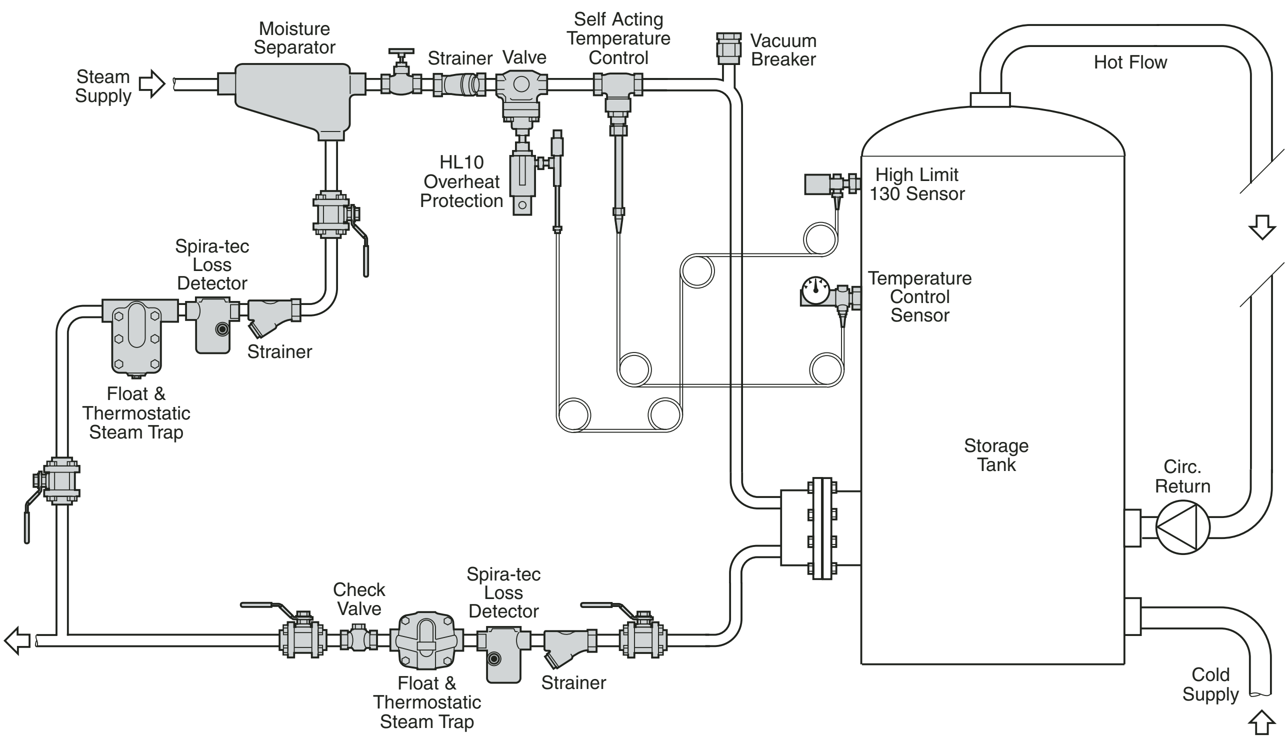
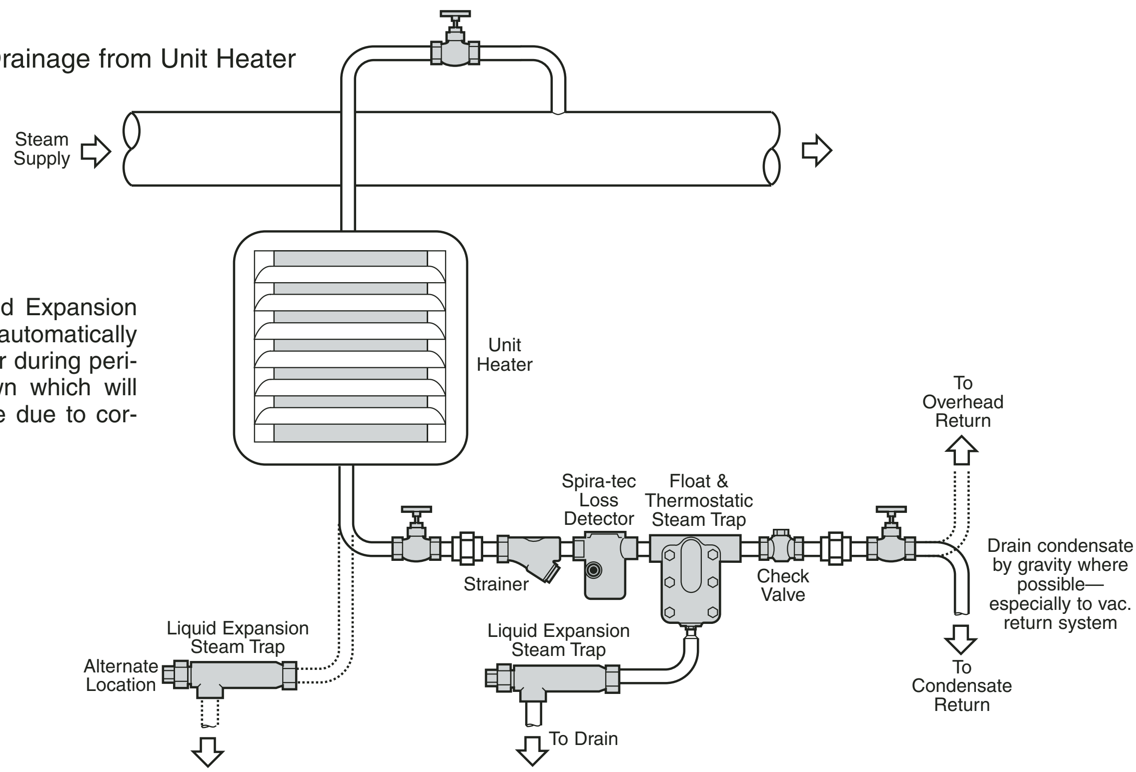


Figure II-30
Condensate Drainage from Unit Heater



Note: The Liquid Expansion Steam Trap will automatically drain unit heater during periods of shutdown which will prevent damage due to corrosion.

Figure II-31

Temperature Control of Warm-up and Running Loads at Storage Tank

A control valve suitably sized to supply the start up load on a tank is often very much oversized for the running load, and this oversizing can lead to erratic control. In such cases, a large control valve may be used to meet the warm up load, arranged to close at a temperature perhaps 2° below the final control temperature. The smaller control valve meets the running load, and the supply is supplemented through the start up valve, only when the capacity of the smaller valve is exceeded.

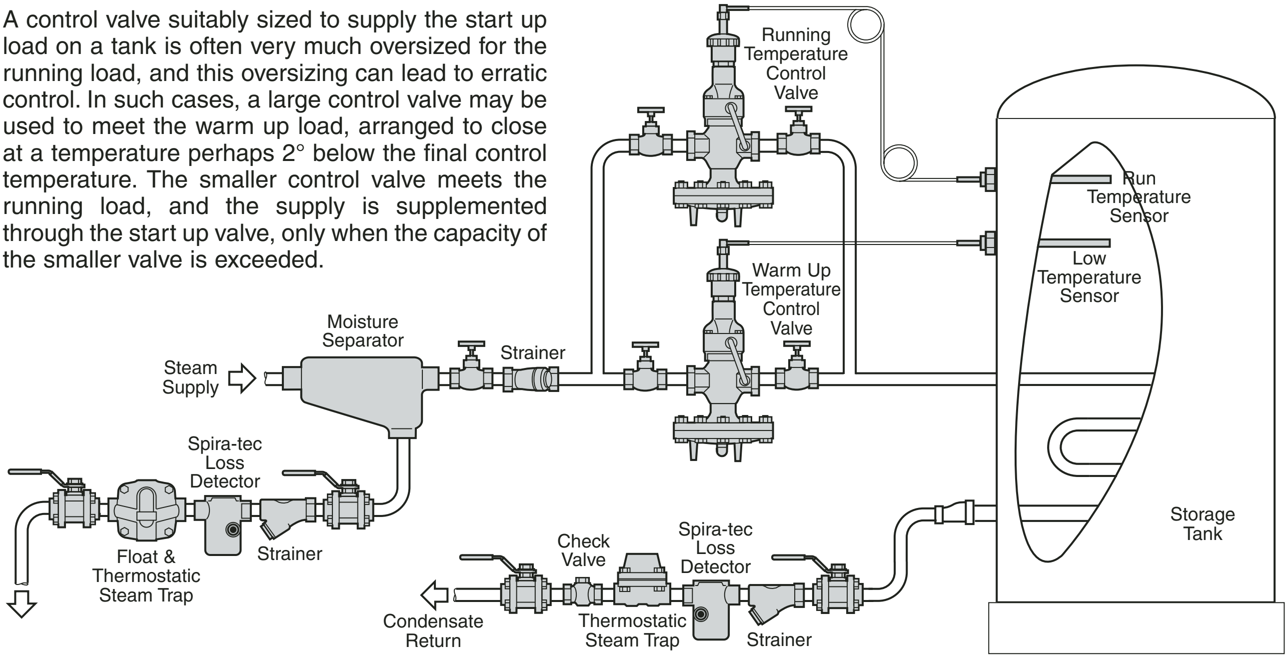
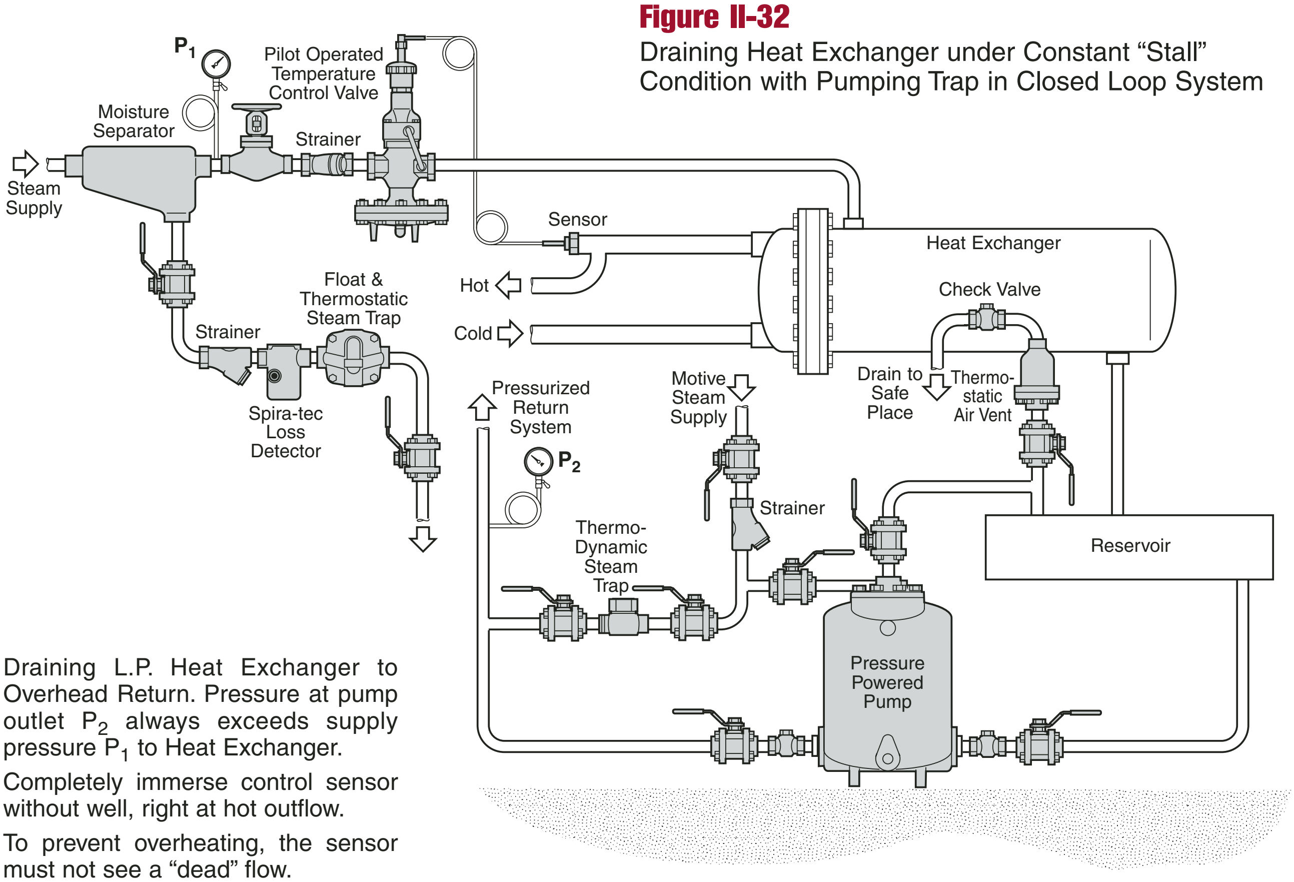


Figure II-32

Draining Heat Exchanger under Constant “Stall” Condition with Pumping Trap in Closed Loop System



Draining L.P. Heat Exchanger to Overhead Return. Pressure at pump outlet P_2 always exceeds supply pressure P_1 to Heat Exchanger.

Completely immerse control sensor without well, right at hot outflow.

To prevent overheating, the sensor must not see a “dead” flow.

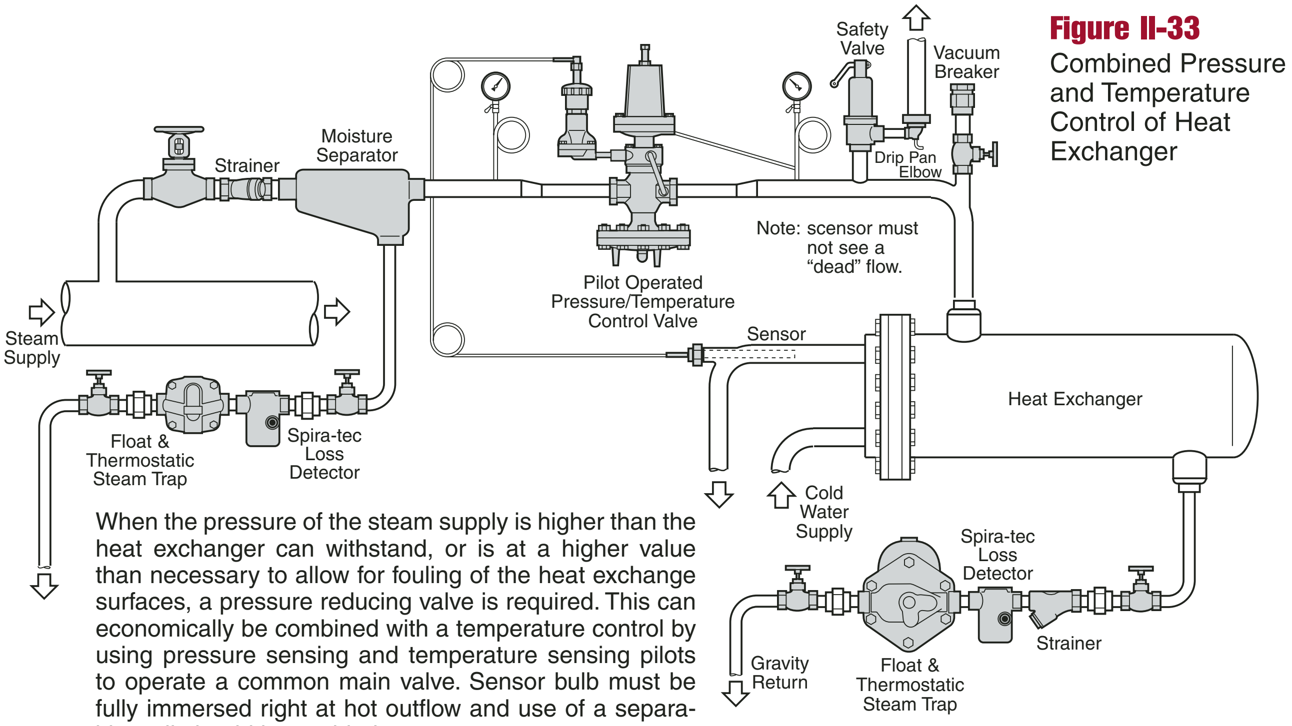


Figure II-33

Combined Pressure and Temperature Control of Heat Exchanger

When the pressure of the steam supply is higher than the heat exchanger can withstand, or is at a higher value than necessary to allow for fouling of the heat exchange surfaces, a pressure reducing valve is required. This can economically be combined with a temperature control by using pressure sensing and temperature sensing pilots to operate a common main valve. Sensor bulb must be fully immersed right at hot outflow and use of a separable well should be avoided.

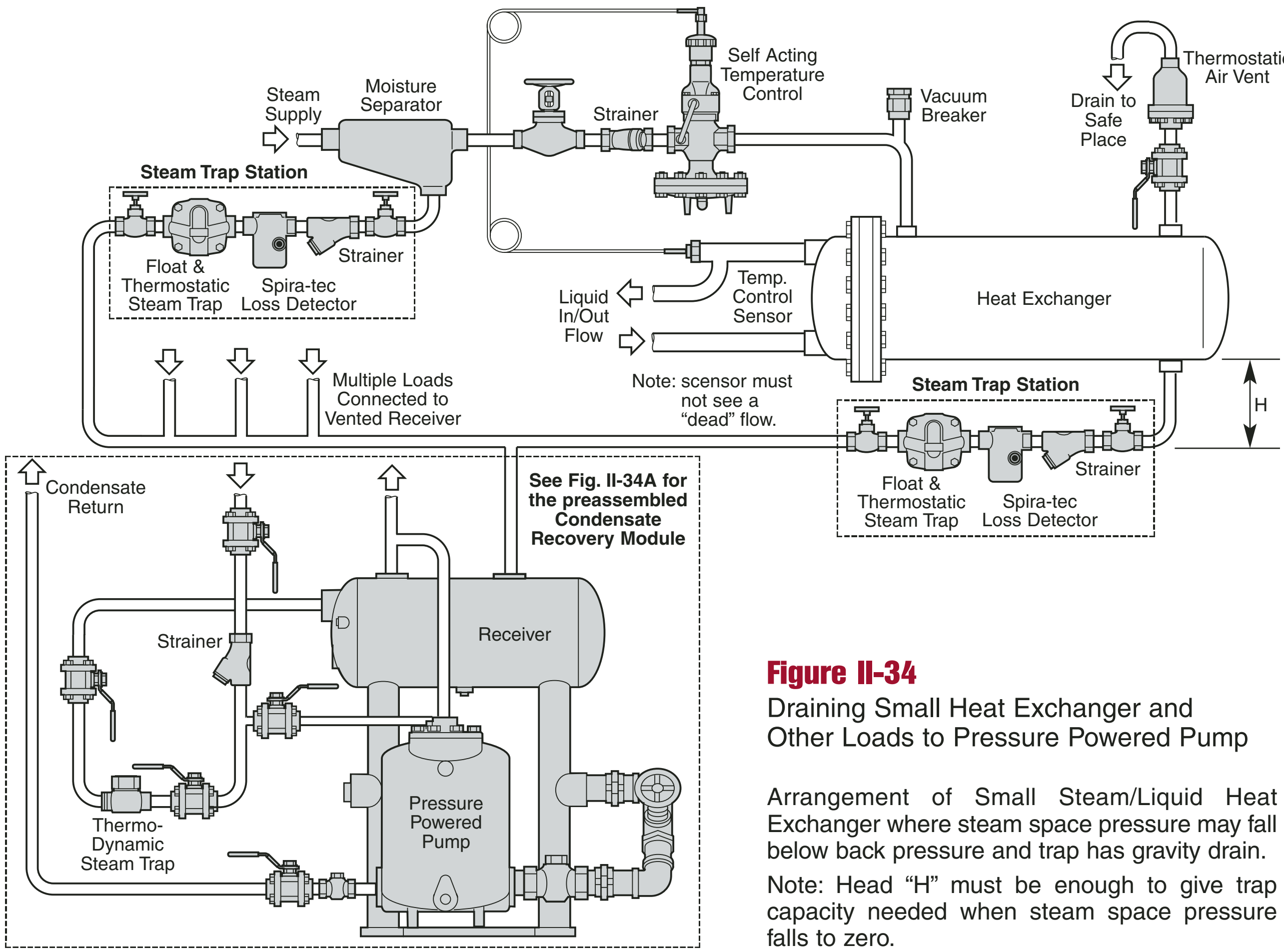


Figure II-34

Draining Small Heat Exchanger and Other Loads to Pressure Powered Pump

Arrangement of Small Steam/Liquid Heat Exchanger where steam space pressure may fall below back pressure and trap has gravity drain.

Note: Head “H” must be enough to give trap capacity needed when steam space pressure falls to zero.

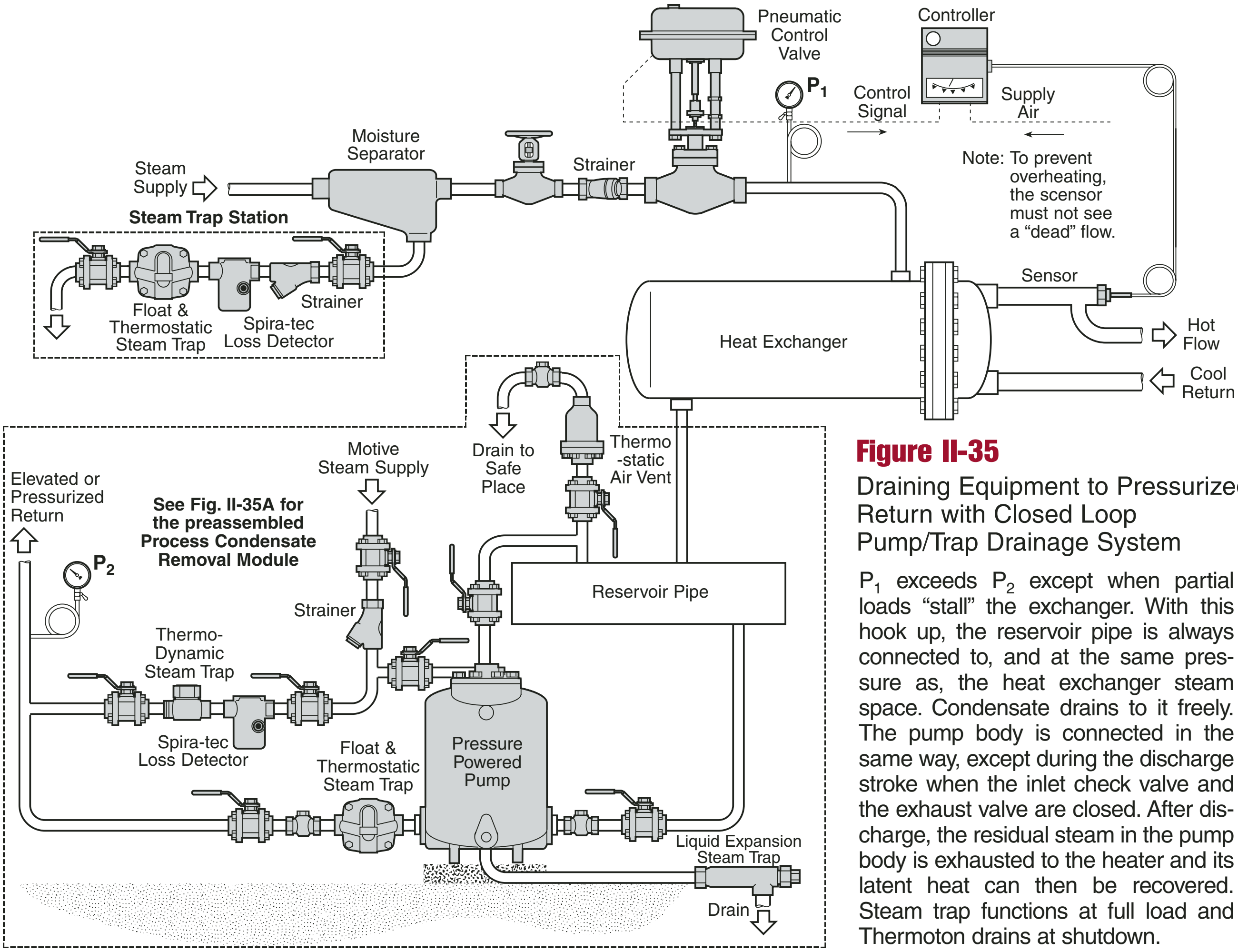


Figure II-35

Draining Equipment to Pressurized Return with Closed Loop Pump/Trap Drainage System

P_1 exceeds P_2 except when partial loads "stall" the exchanger. With this hook up, the reservoir pipe is always connected to, and at the same pressure as, the heat exchanger steam space. Condensate drains to it freely. The pump body is connected in the same way, except during the discharge stroke when the inlet check valve and the exhaust valve are closed. After discharge, the residual steam in the pump body is exhausted to the heater and its latent heat can then be recovered. Steam trap functions at full load and Thermoton drains at shutdown.

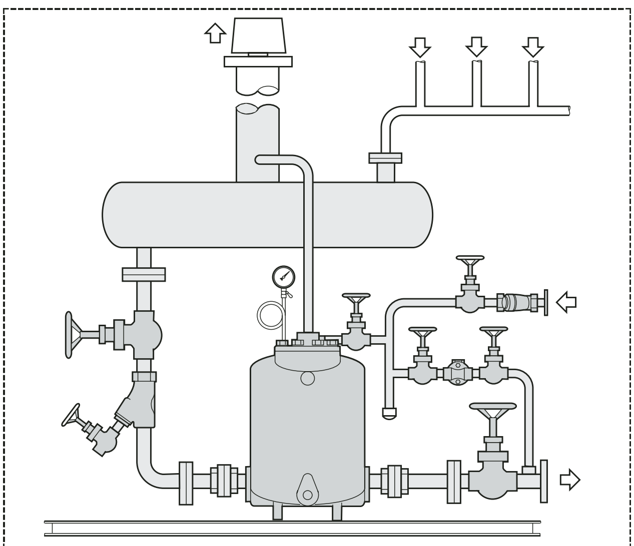


Figure II-34A

Condensate Recovery Module

A preassembled modular pumping system can be used to recover and reuse the condensate.

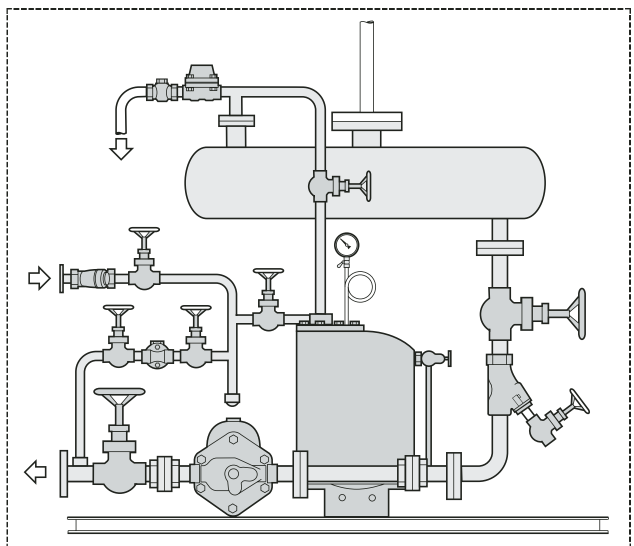


Figure II-35A

Process Condensate Removal Module

A preassembled modular will remove condensate from the heat exchanger under all operating conditions.

Figure II-36
Low Pressure Steam Absorption Chiller

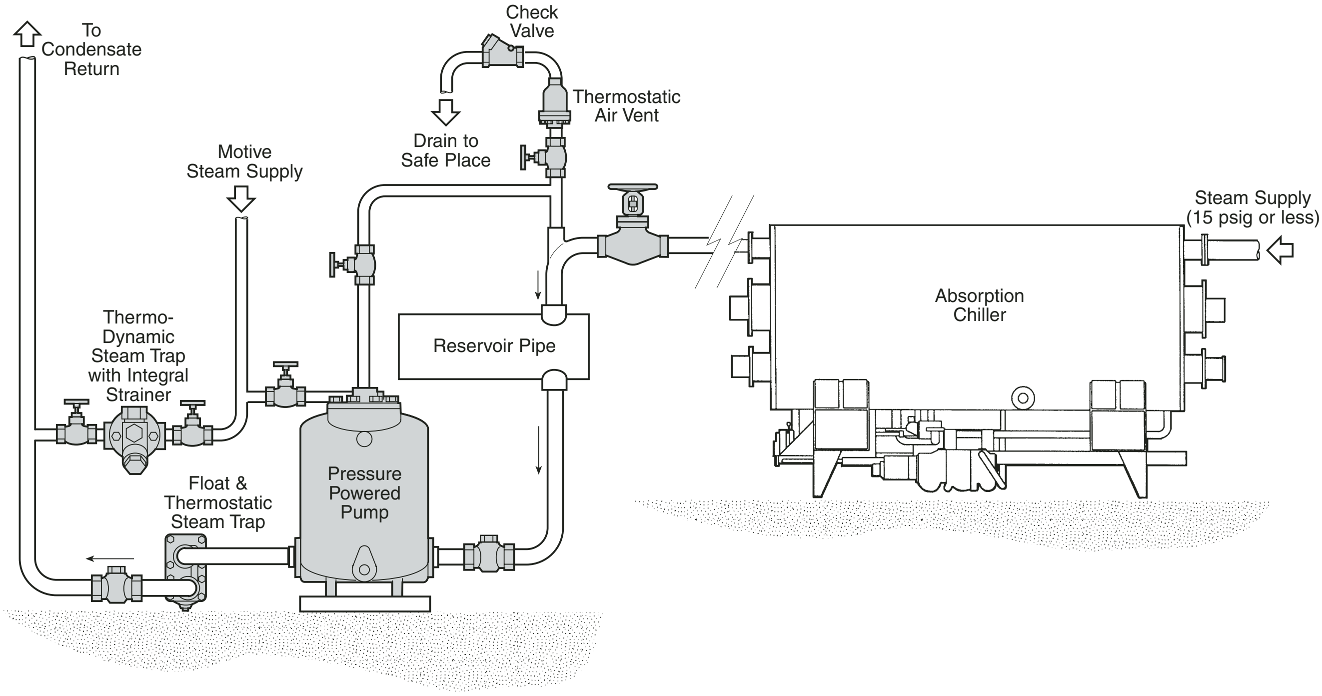
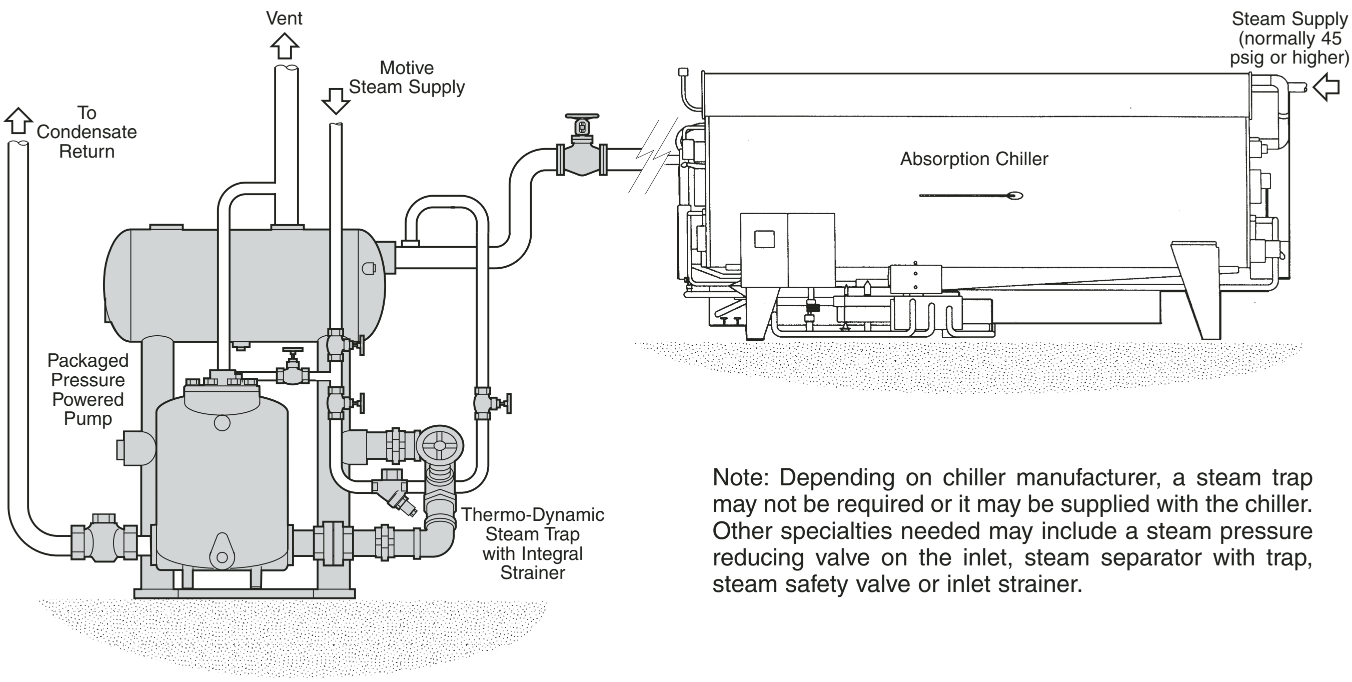


Figure II-37
High Pressure Steam Absorption Chiller



Note: Depending on chiller manufacturer, a steam trap may not be required or it may be supplied with the chiller. Other specialties needed may include a steam pressure reducing valve on the inlet, steam separator with trap, steam safety valve or inlet strainer.

Figure II-38 Automatic Control of Batch Processor with Electrical Time Sequence Programmer

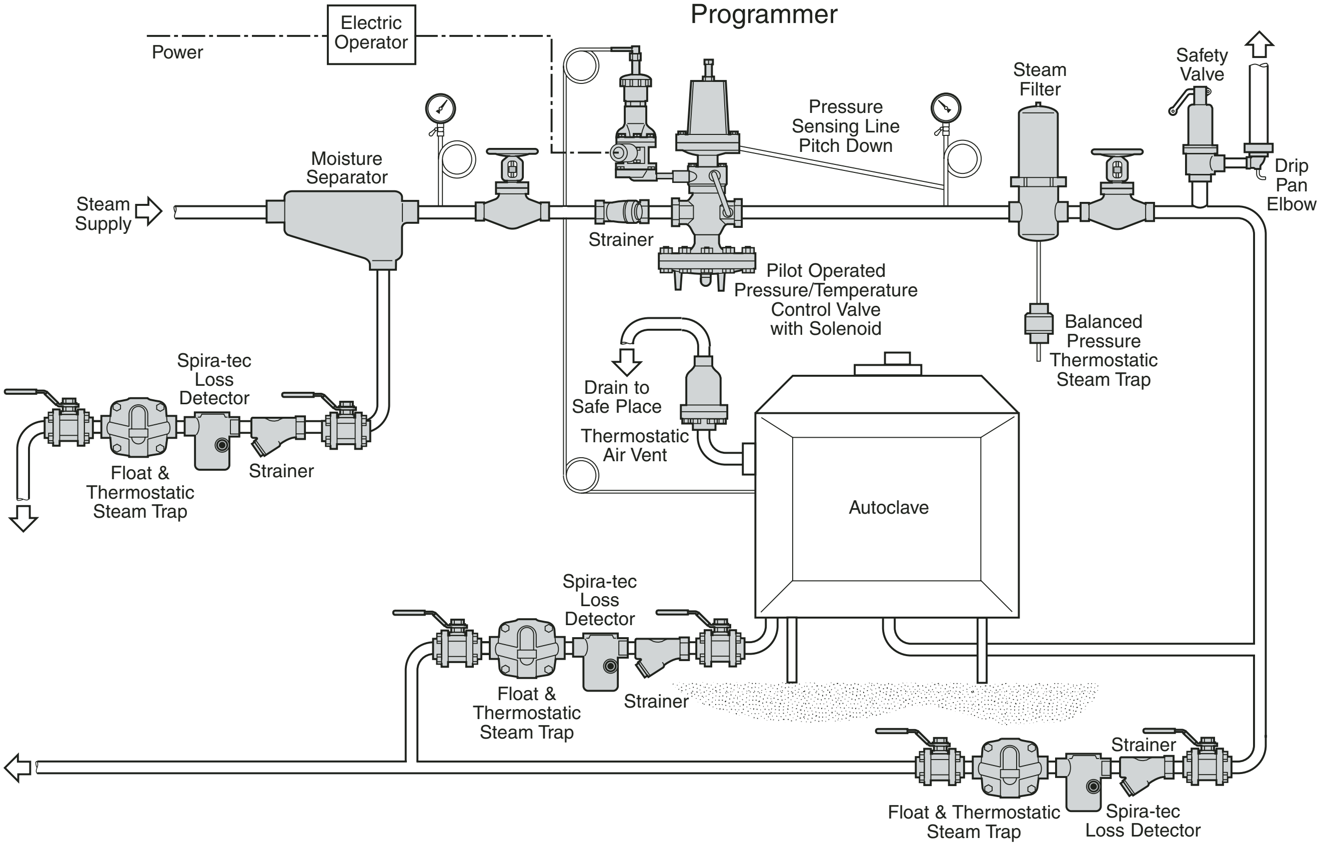
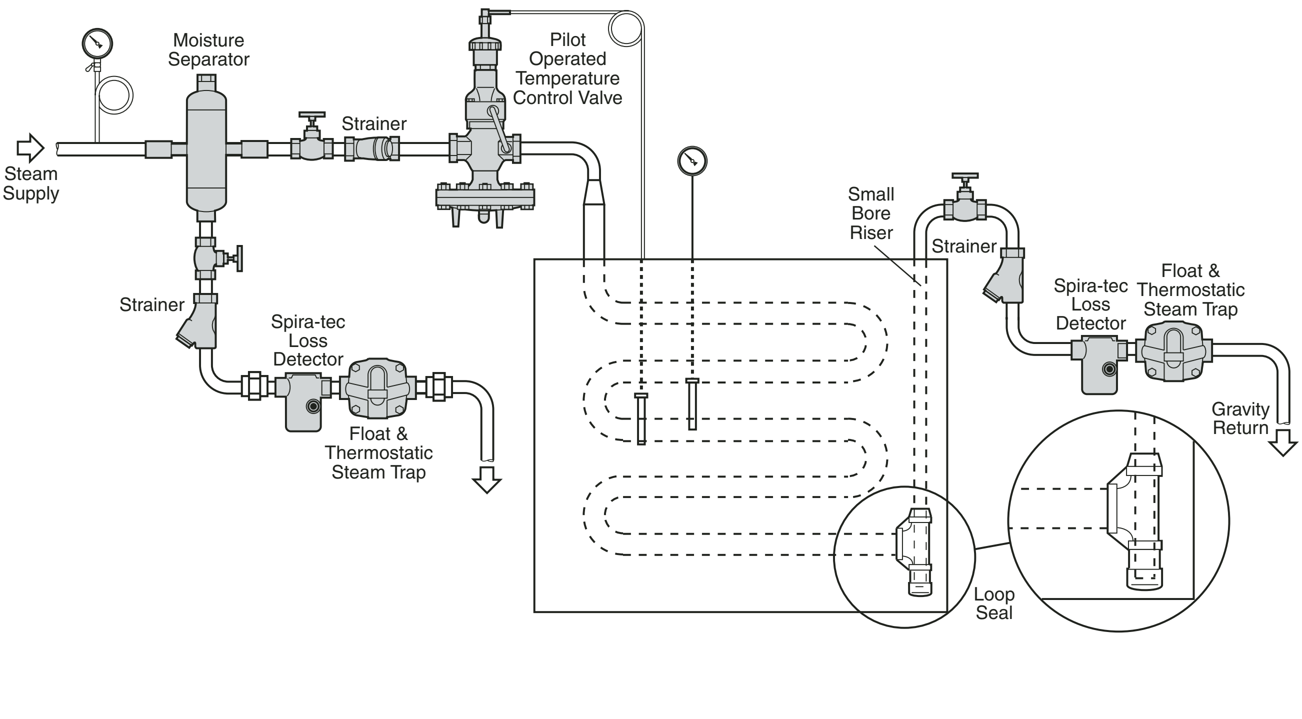


Figure II-39
Controlling Temperature of Open Tank for Plating, Dyeing of Process Work



HOOK-UP DIAGRAMS

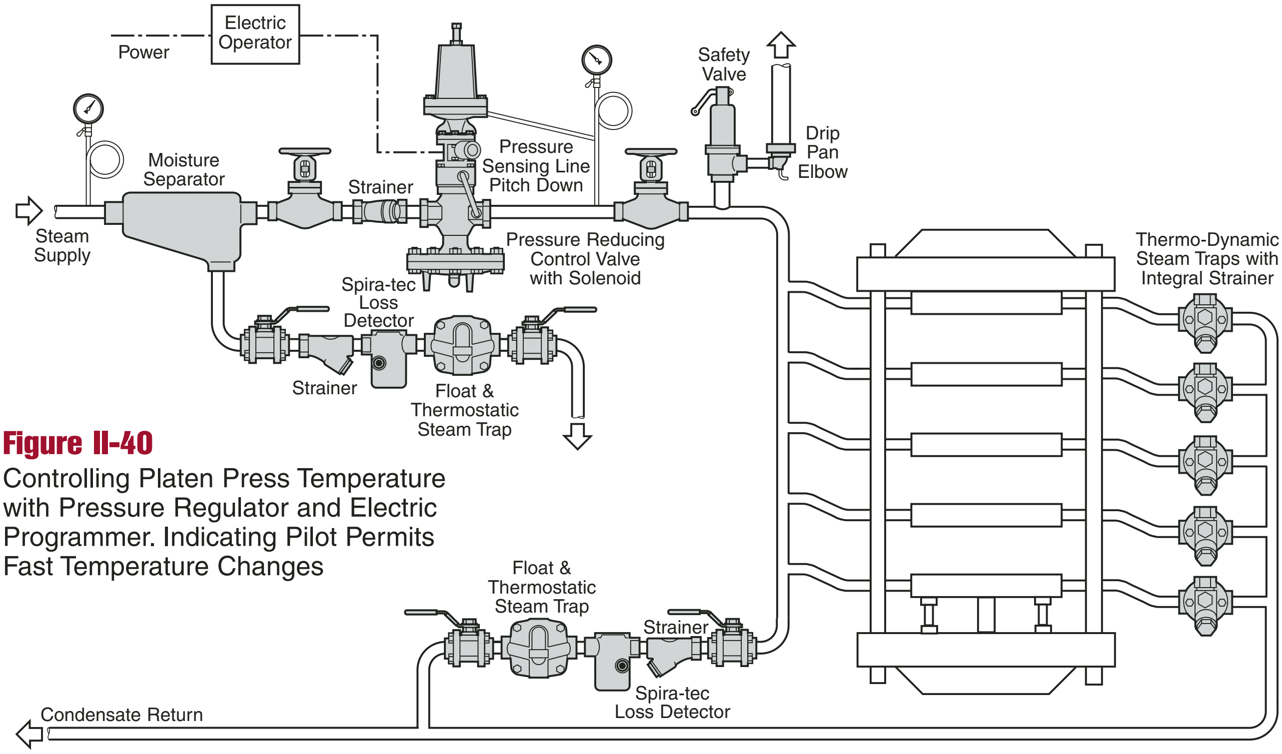
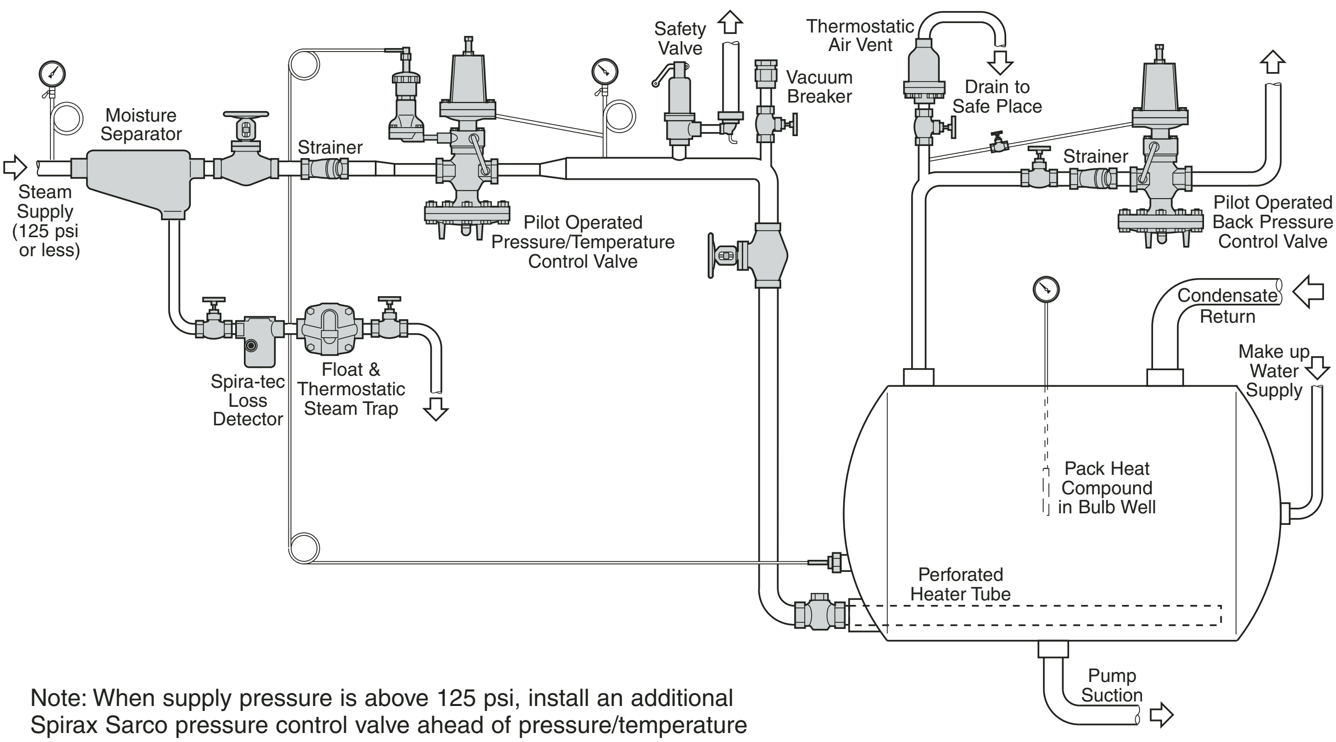


Figure II-40

Controlling Platen Press Temperature with Pressure Regulator and Electric Programmer. Indicating Pilot Permits Fast Temperature Changes

Figure II-41

Controlling Temperature of Pressurized Boiler Feed Water Tank



Note: When supply pressure is above 125 psi, install an additional Spirax Sarco pressure control valve ahead of pressure/temperature control valve to reduce pressure to 125 psi.

Figure II-42
Controlling Temperature of Vented Boiler Feed Water Tank

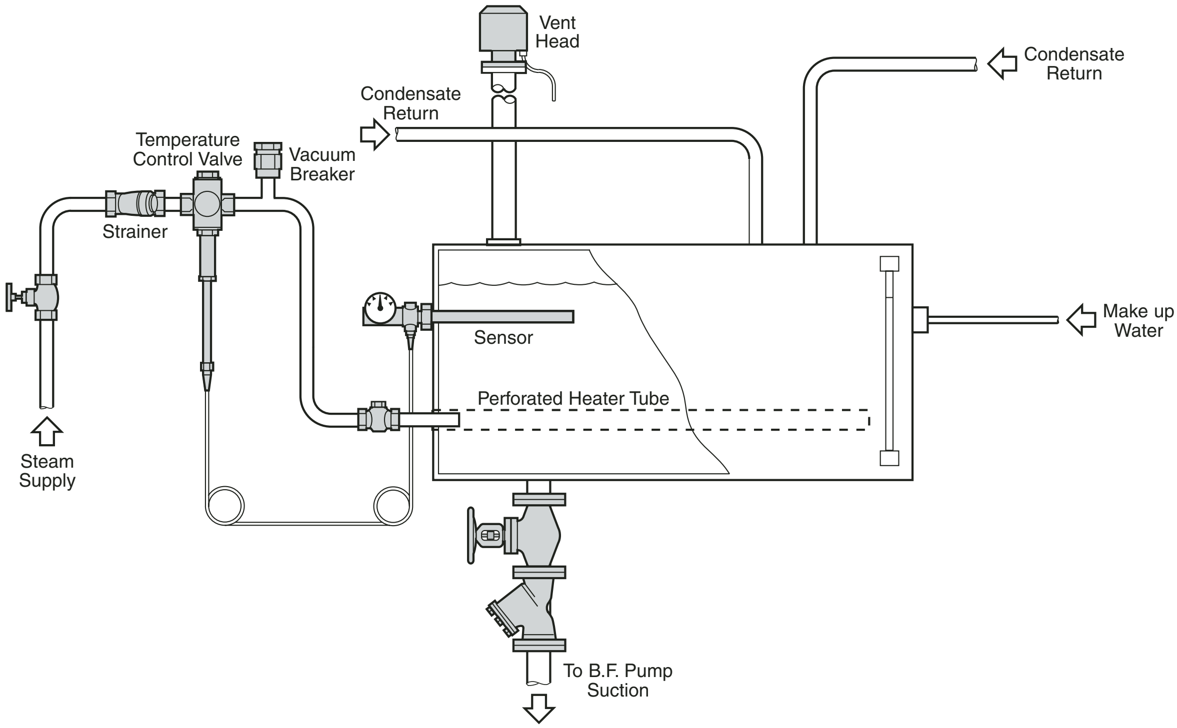
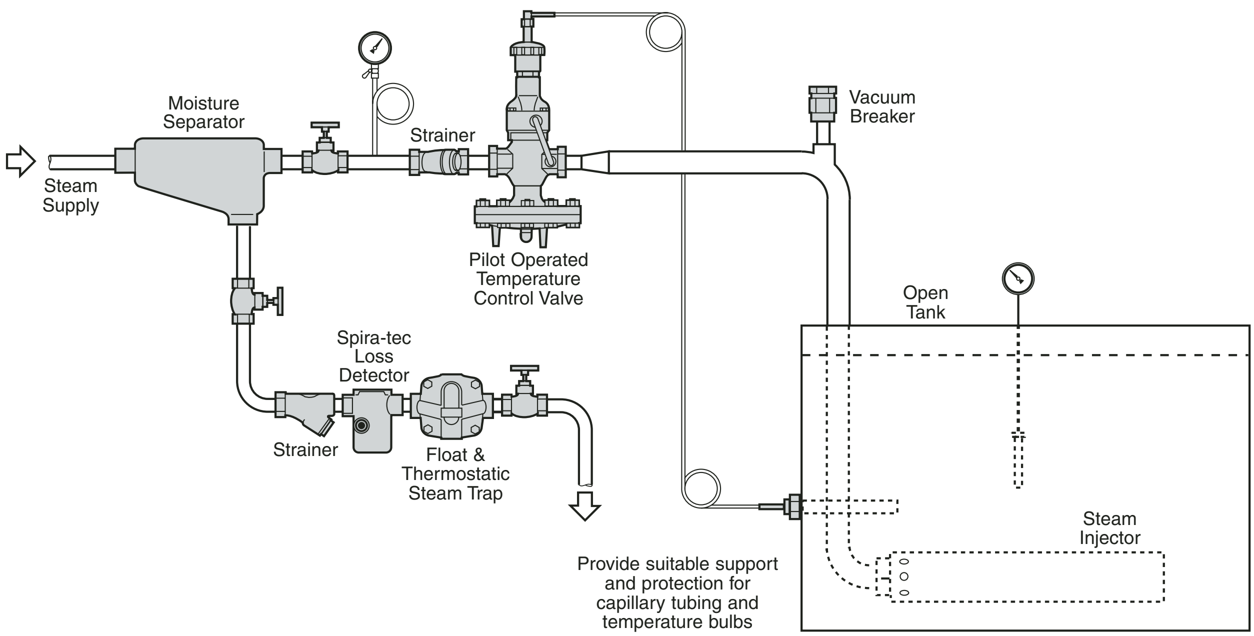


Figure II-43
Controlling Temperature of Large Open Tank Heated by Direct Steam Injection



HOOK-UP DIAGRAMS

Figure II-44
Controlling Temperature of Small Open Tank, Heated by Direct Steam Injection

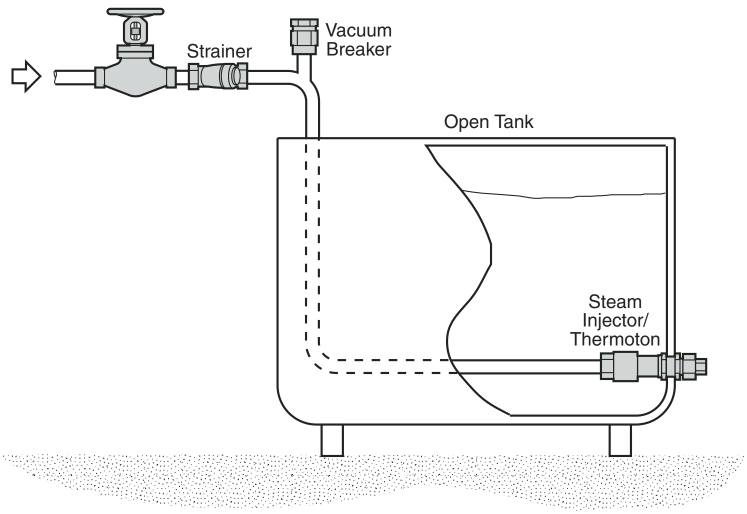


Figure II-45
Controlling Temperature of Water Supplied to Spray Nozzles of Egg Washing Machine

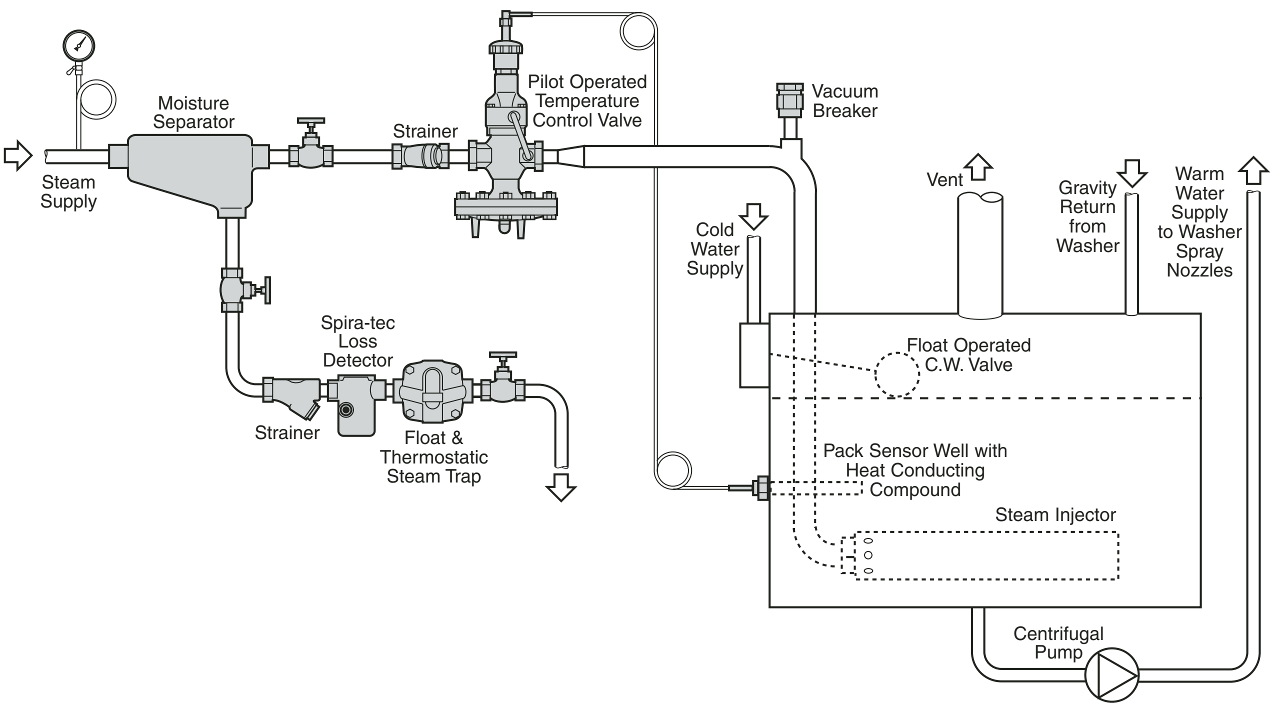


Figure II-46

Controlling Temperature of Greenhouse or Other Similar Buildings

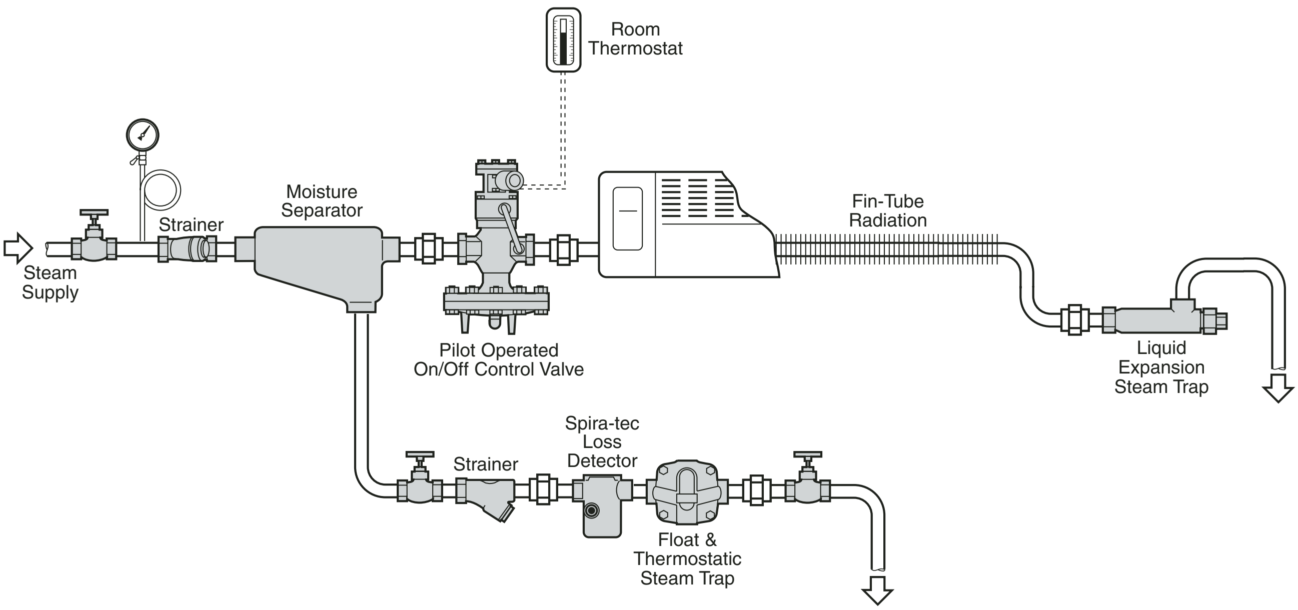


Figure II-47

Steam Radiator

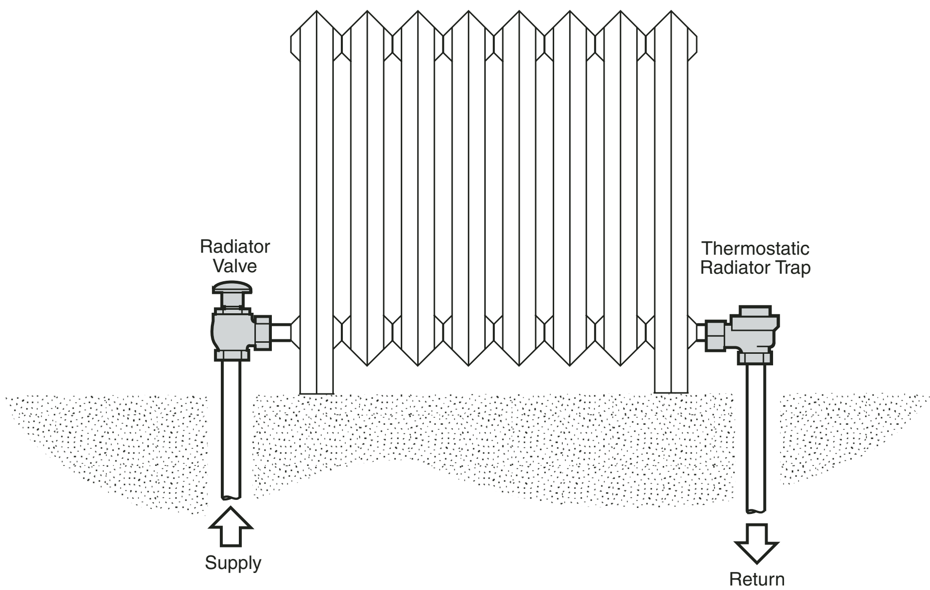
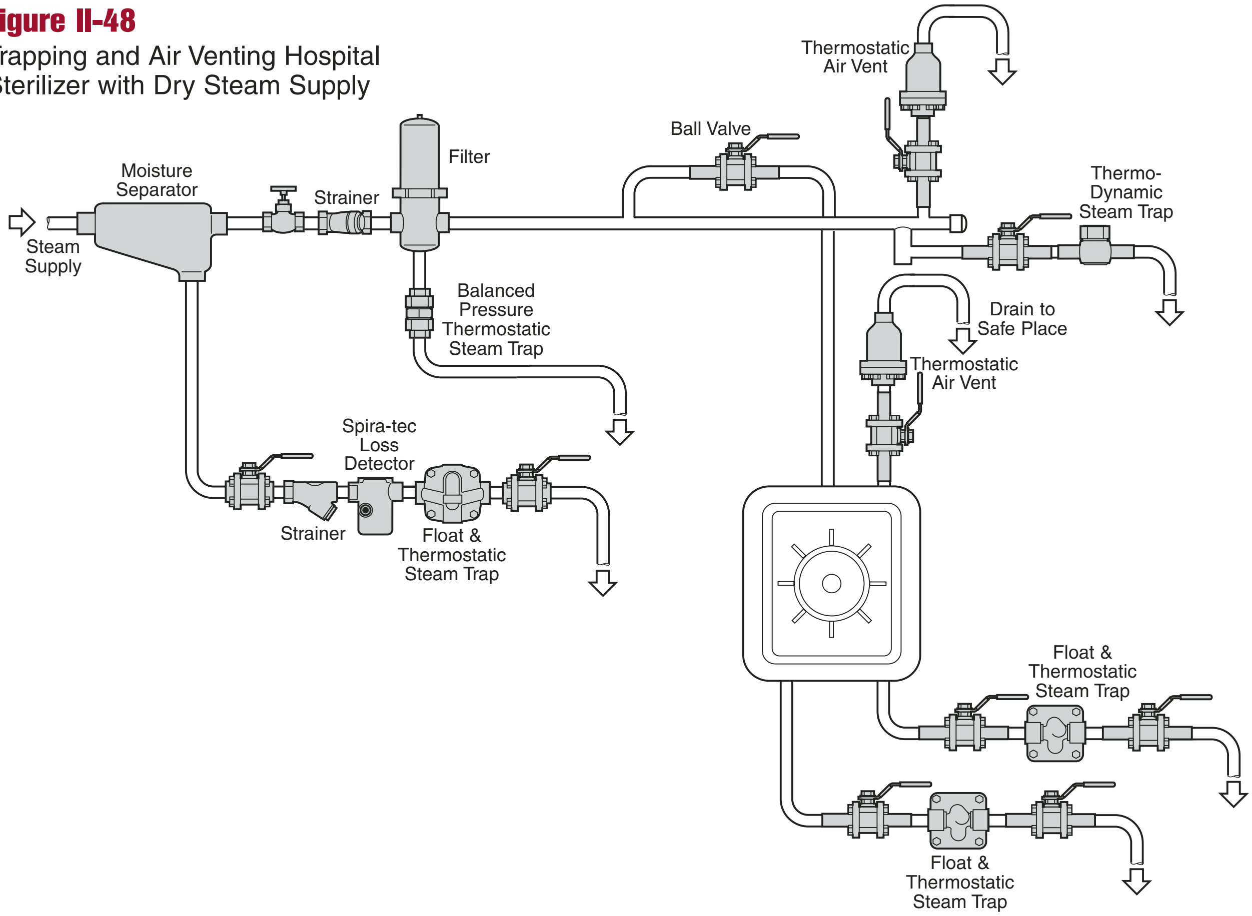


Figure II-48

Trapping and Air Venting Hospital Sterilizer with Dry Steam Supply



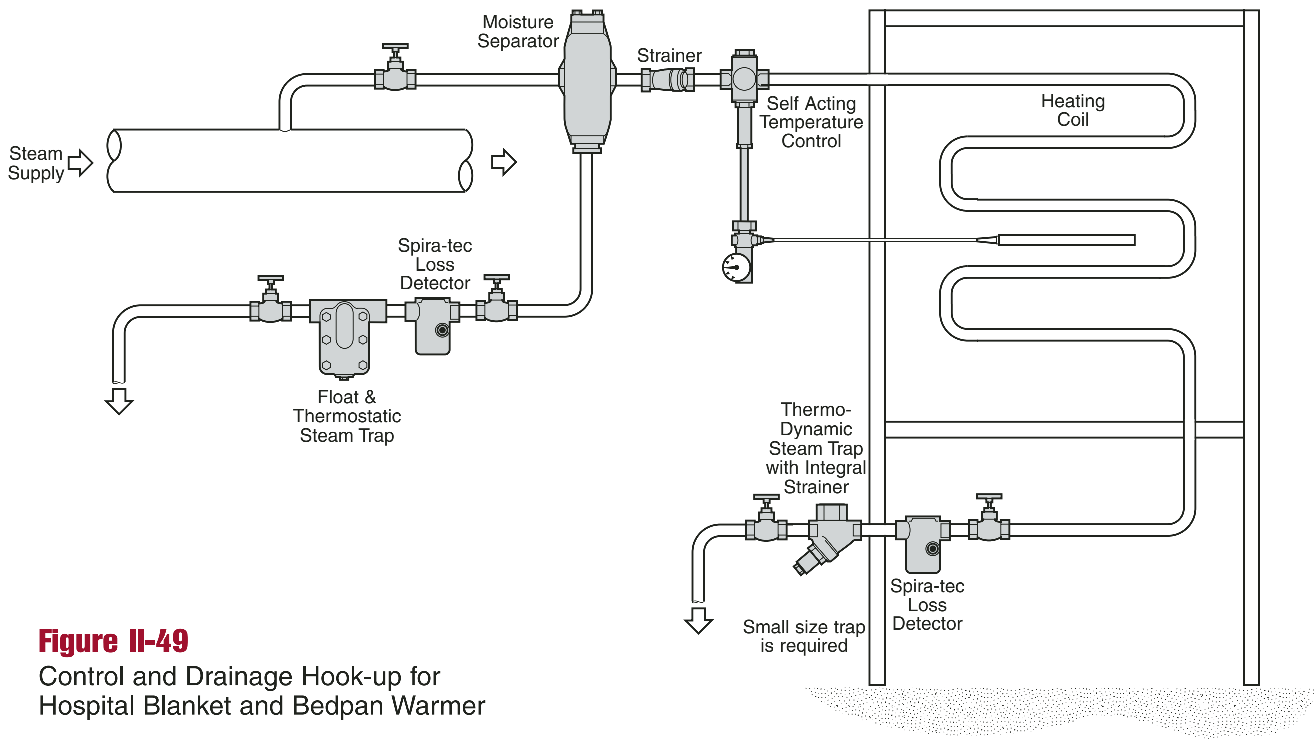


Figure II-49

Control and Drainage Hook-up for Hospital Blanket and Bedpan Warmer

Figure II-50
Trapping Small Utensil Sterilizer

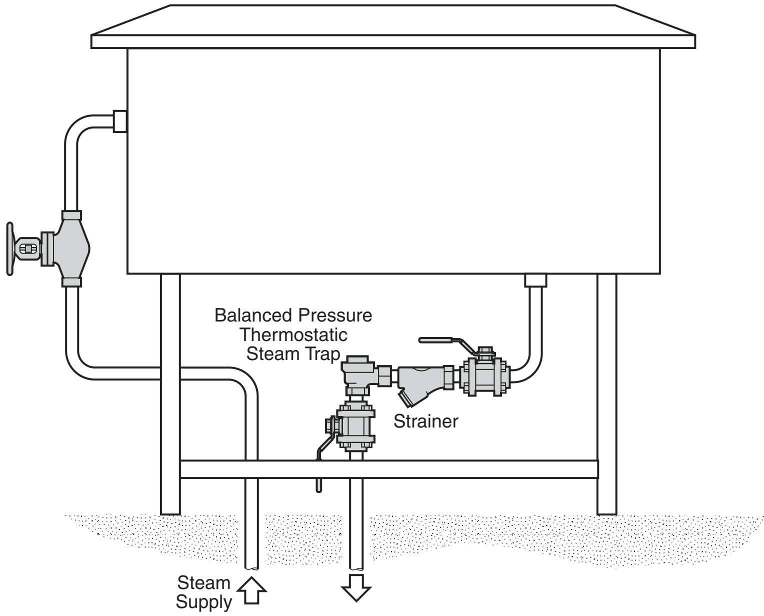


Figure II-51
Condensate Drainage from
Hospital Mattress Disinfector

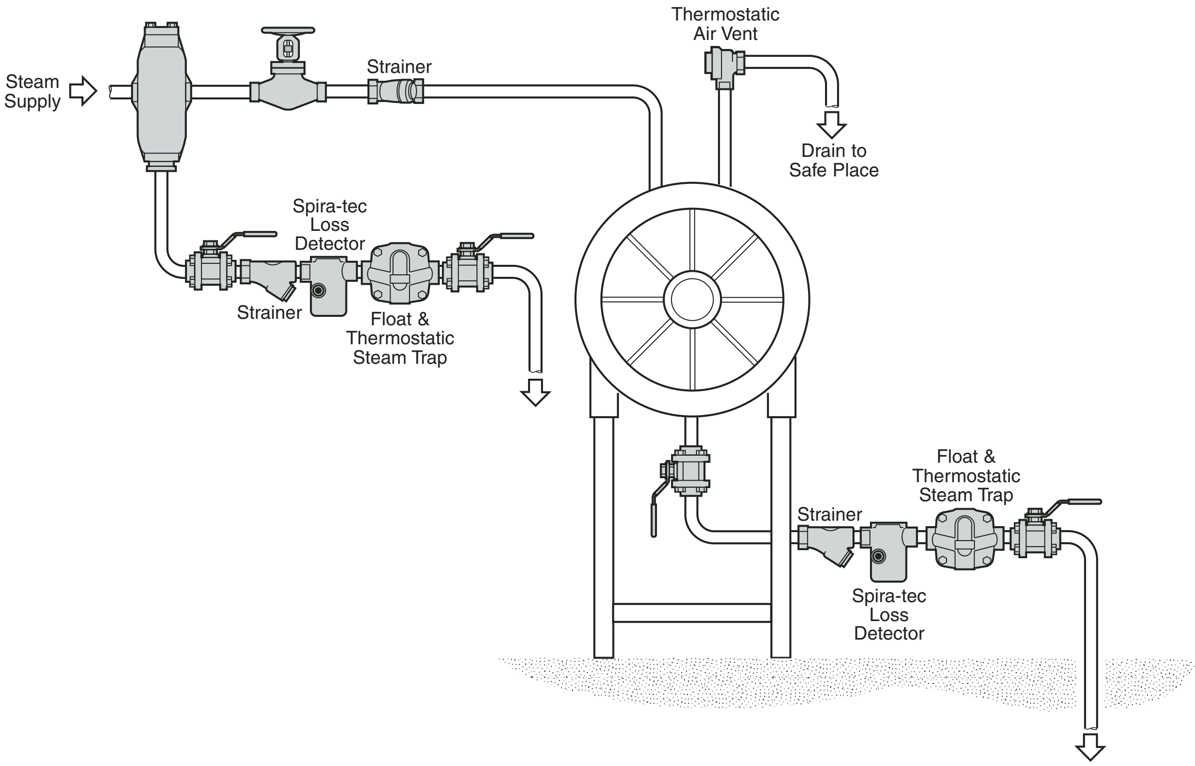


Figure II-52
Float & Thermostatic Trap
Freeze Resistant Hook-up

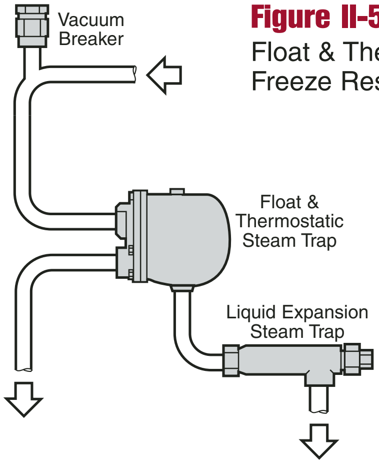


Figure II-53
Thermoton Controlling Temperature
of Large Storage Tank

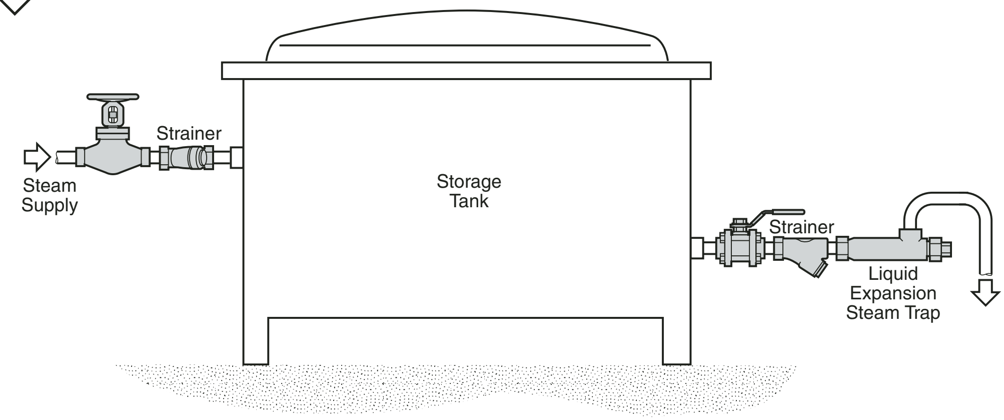


Figure II-54
Equipment Drained with Permanent Connector Thermo-Dynamic Steam Traps
that fit into both Horizontal and Vertical Pipework

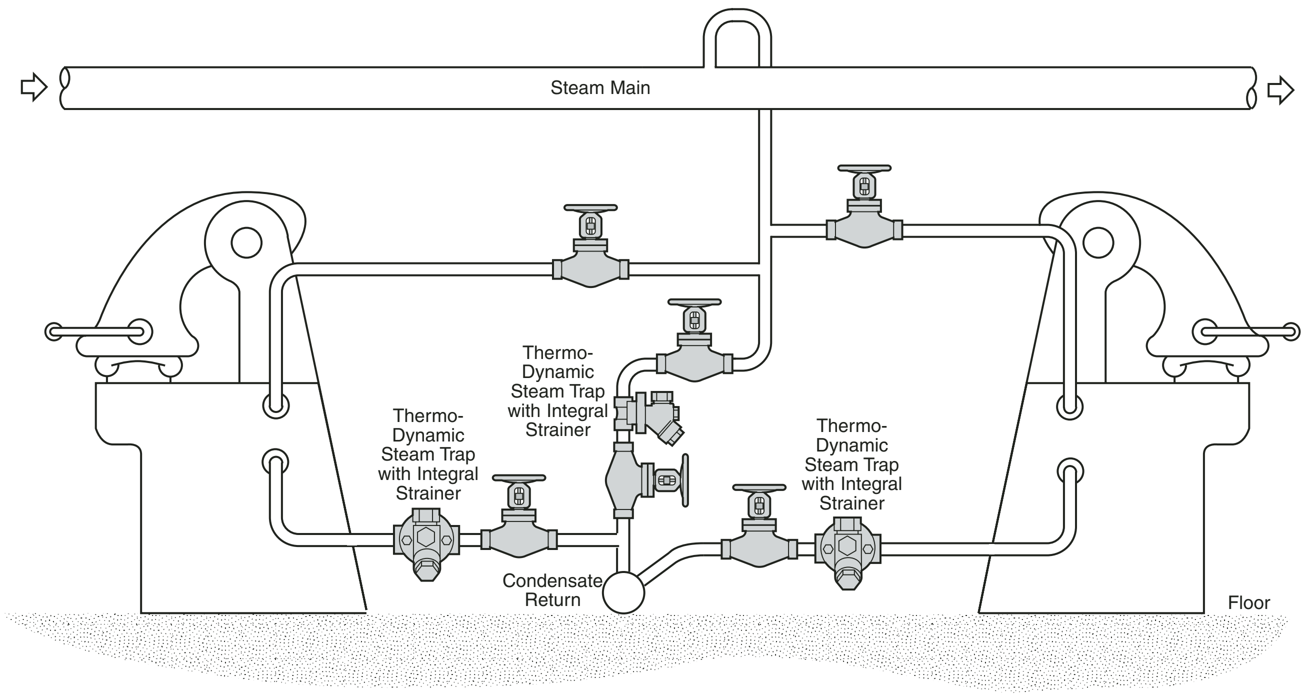
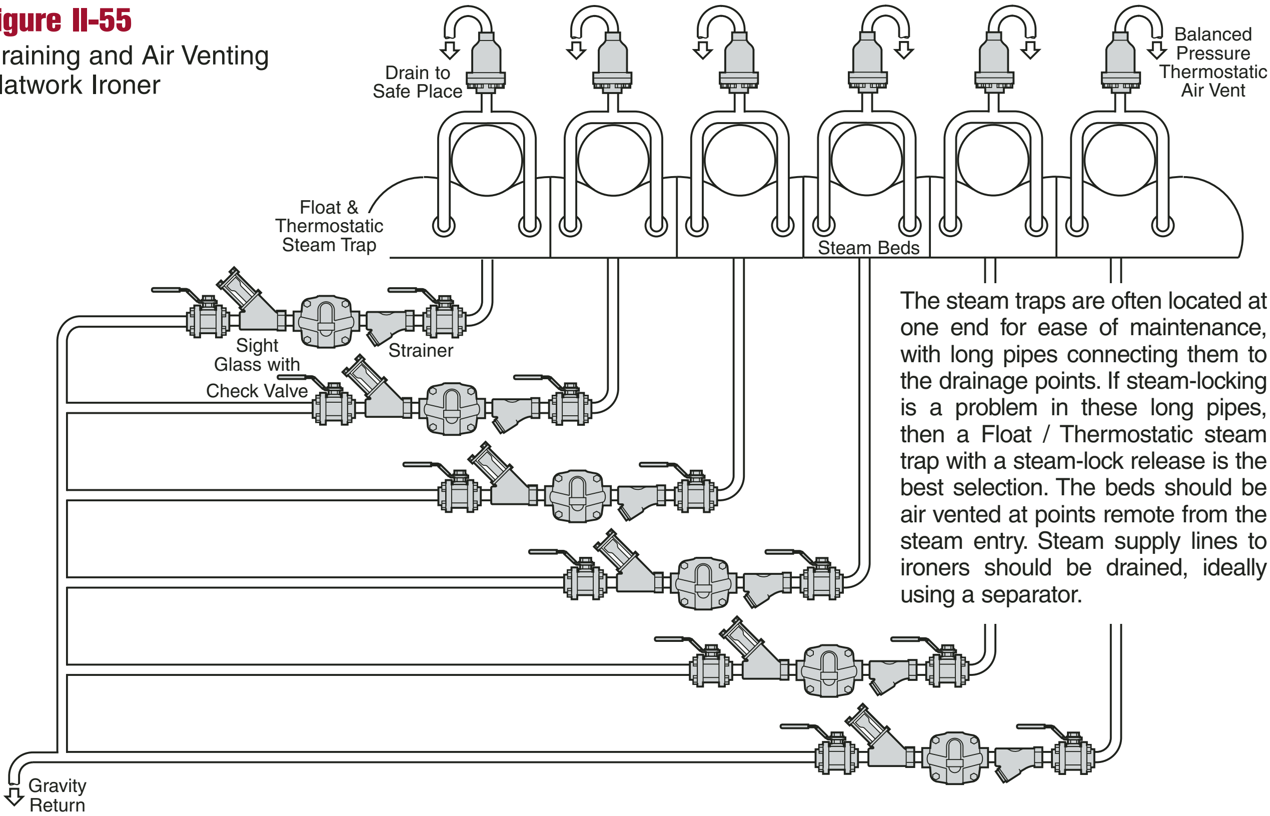


Figure II-55
Draining and Air Venting Flatwork Ironer



The steam traps are often located at one end for ease of maintenance, with long pipes connecting them to the drainage points. If steam-locking is a problem in these long pipes, then a Float / Thermostatic steam trap with a steam-lock release is the best selection. The beds should be air vented at points remote from the steam entry. Steam supply lines to ironers should be drained, ideally using a separator.

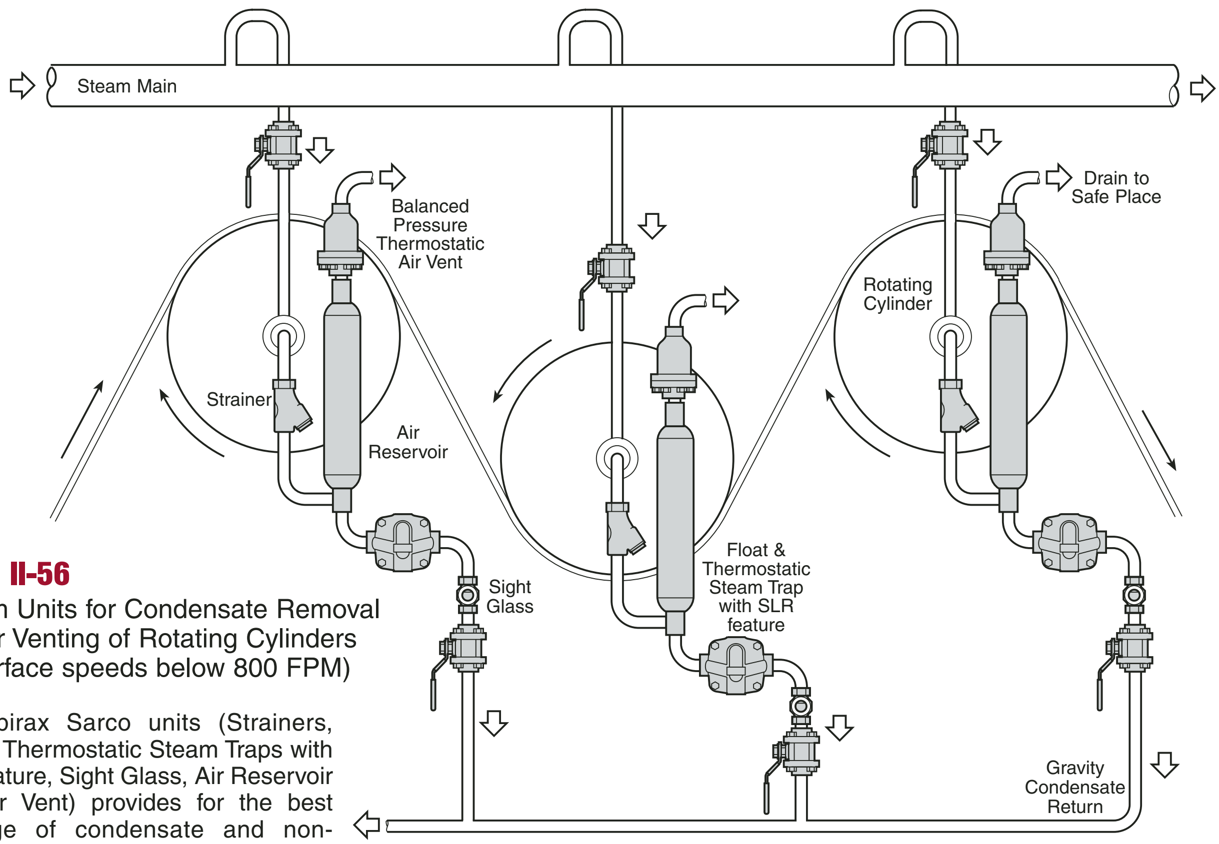
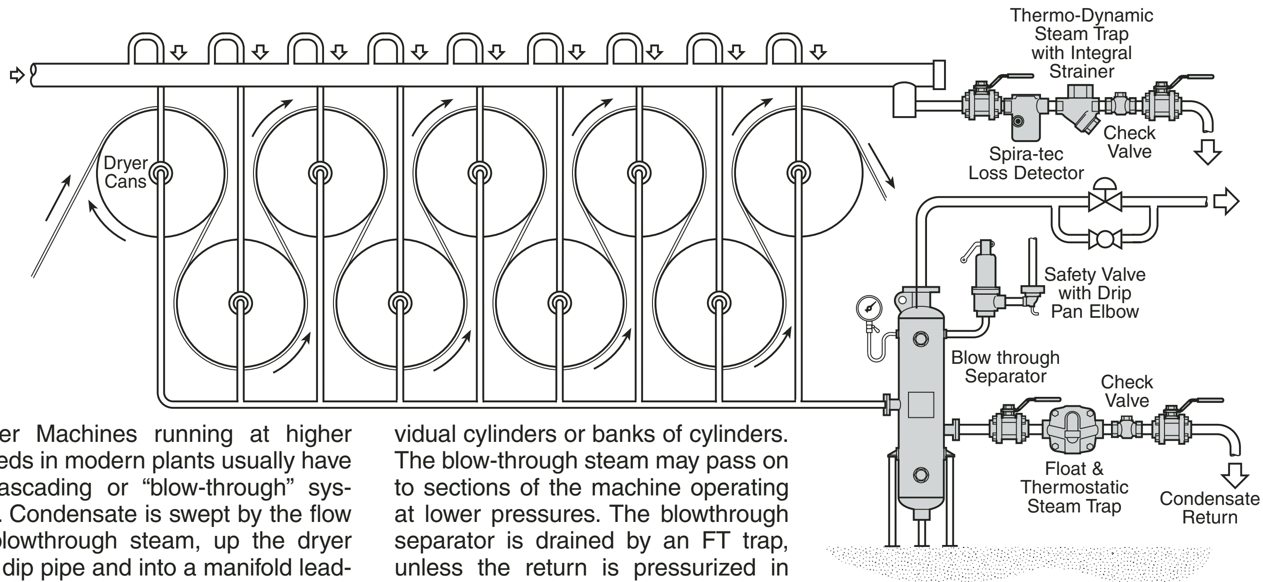


Figure II-56
System Units for Condensate Removal and Air Venting of Rotating Cylinders (for surface speeds below 800 FPM)

The Spirax Sarco units (Strainers, Float & Thermostatic Steam Traps with SLR feature, Sight Glass, Air Reservoir with Air Vent) provides for the best drainage of condensate and non-condensibles from rotating cylinders.

Figure II-57 Draining High Speed Paper Machine using Cascading or "Blow-through" Systems



Paper Machines running at higher speeds in modern plants usually have a cascading or "blow-through" system. Condensate is swept by the flow of blowthrough steam, up the dryer can dip pipe and into a manifold leading to a blowthrough separator. Control of the blowthrough steam may be at the separator outlet or on individual cylinders or banks of cylinders. The blow-through steam may pass on to sections of the machine operating at lower pressures. The blowthrough separator is drained by an FT trap, unless the return is pressurized in which case a closed loop combination pressure powered pump/trap is required.

Figure II-58 Draining High Speed Paper Machine using "Thermal-compressor" or Reused Steam Systems

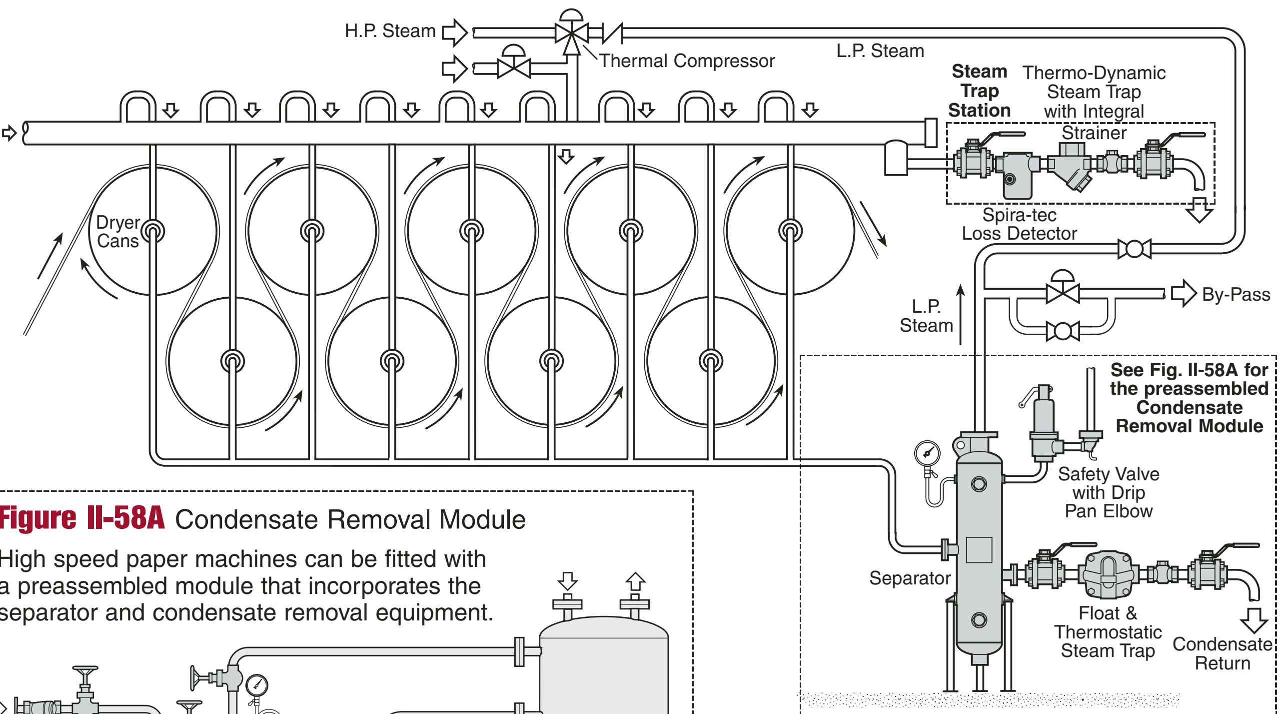
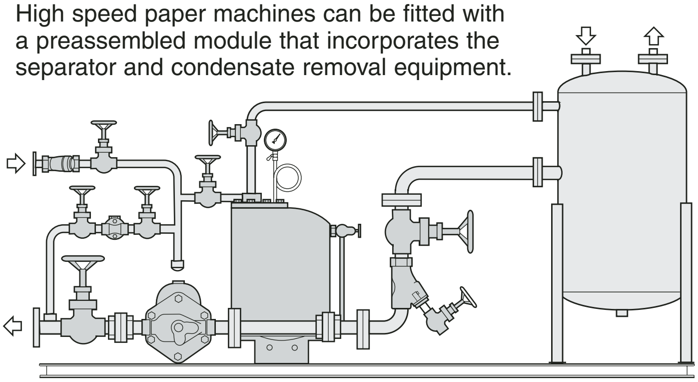


Figure II-58A Condensate Removal Module

High speed paper machines can be fitted with a preassembled module that incorporates the separator and condensate removal equipment.



Modern paper machines also use "Thermal-compressor" systems. Condensate is swept by the flow of blow-through steam up the dryer can dip pipe and into a manifold leading to a separator. The L.P. steam from the separator is then reused through an H.P. steam driven thermal compressor or by-passed. The separator is drained like Fig. II-57 or engineered systems package, Fig. II-58A.

Figure II-59
Air Venting and Condensate Drainage at Jacketed Kettle

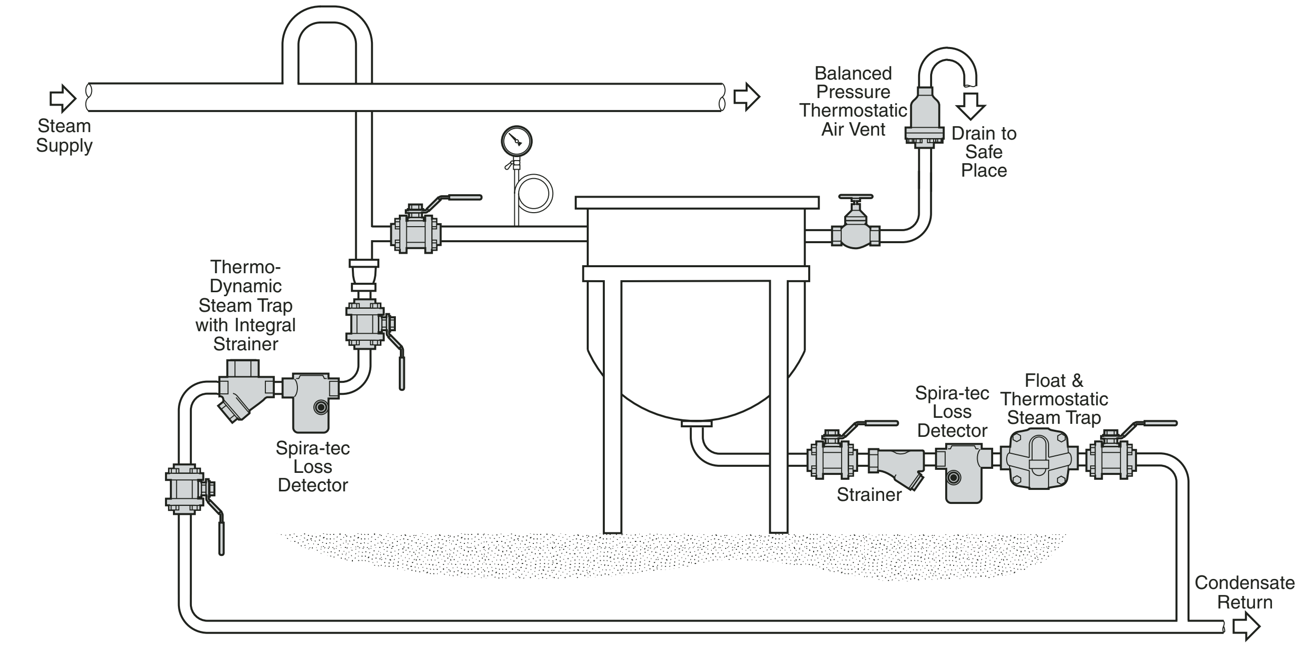


Figure II-60
Draining Tire Mold

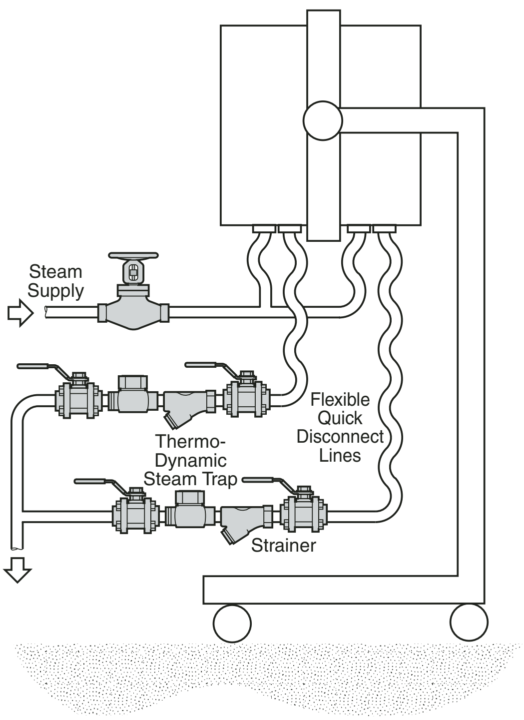


Figure II-61
Steam Trapping High Pressure Coil (up to 600 psig)

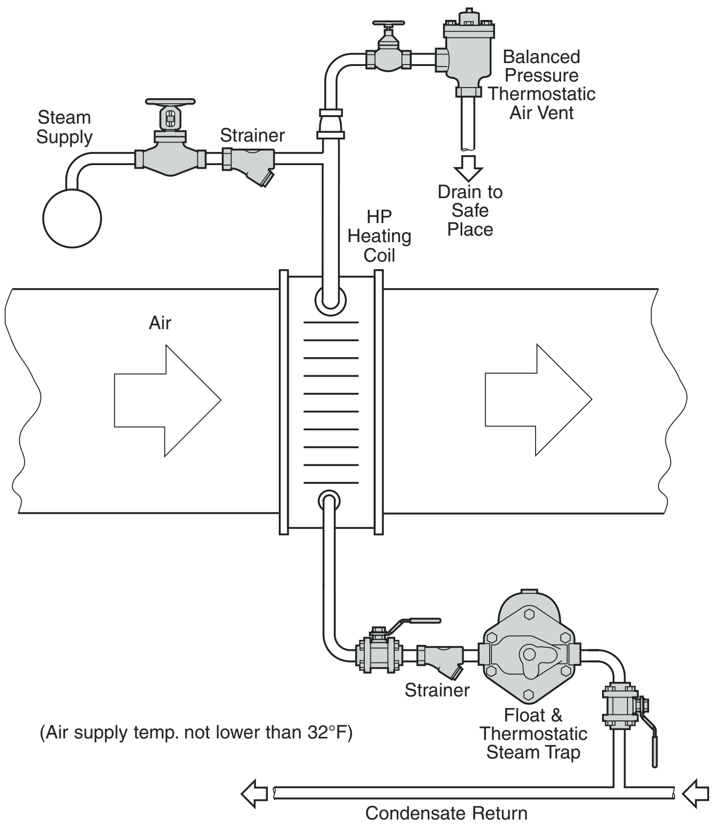


Figure II-62
Draining High Pressure Reboiler

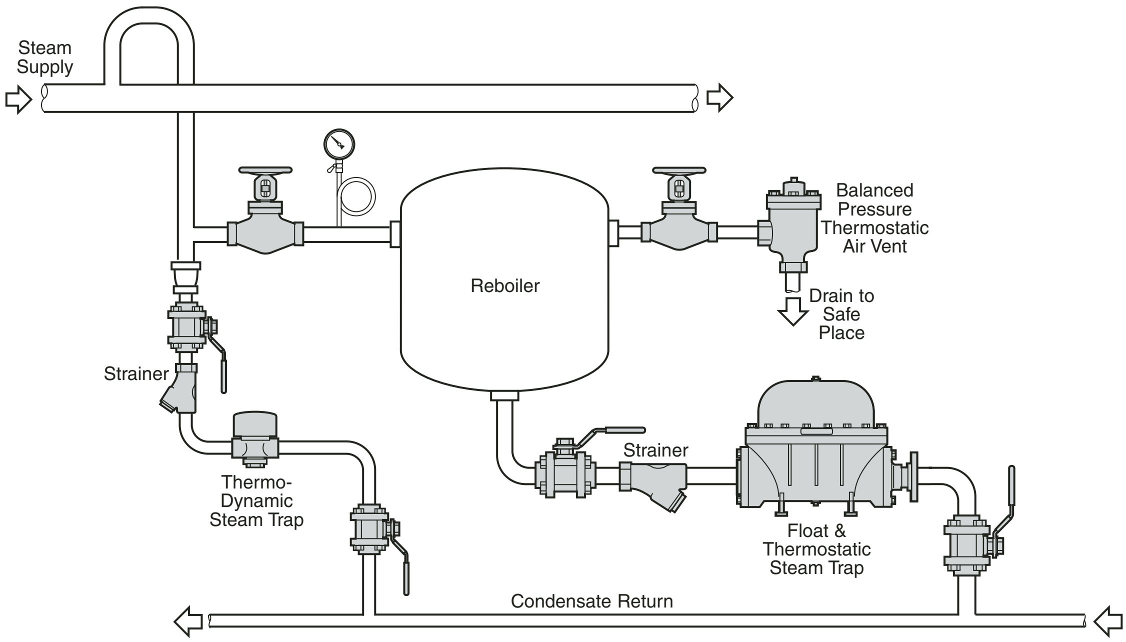
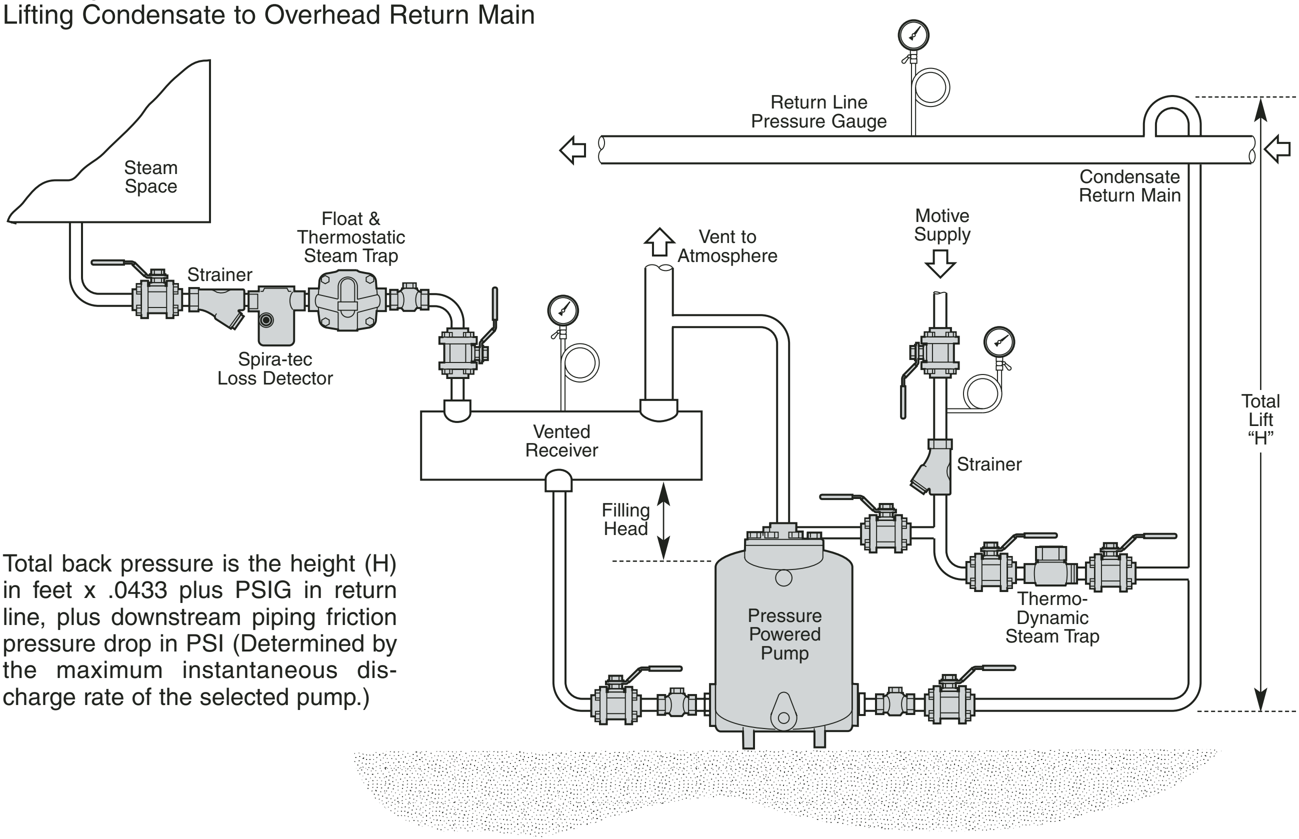


Figure II-63
Draining Condensate to Vented Receiver and Lifting Condensate to Overhead Return Main



Total back pressure is the height (H) in feet x .0433 plus PSIG in return line, plus downstream piping friction pressure drop in PSI (Determined by the maximum instantaneous discharge rate of the selected pump.)

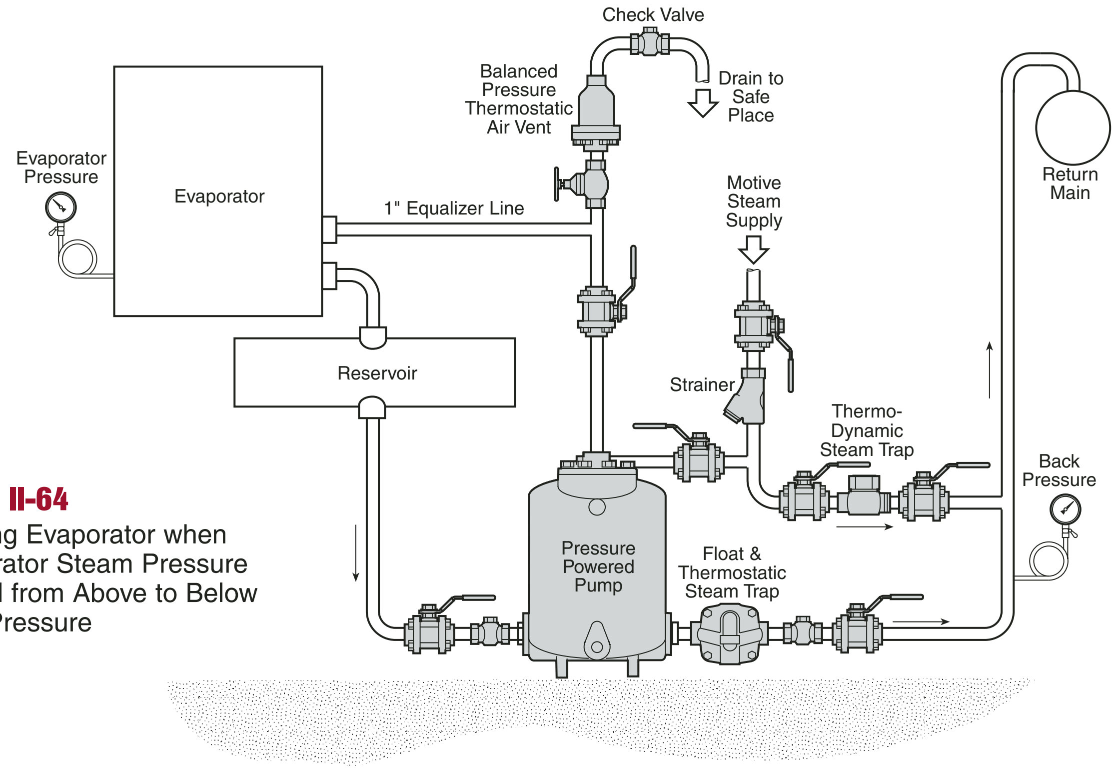


Figure II-64

Draining Evaporator when Evaporator Steam Pressure can fall from Above to Below Back Pressure

Figure II-65

Draining Condensate from vacuum Space to Return Main or Atmosphere Drain

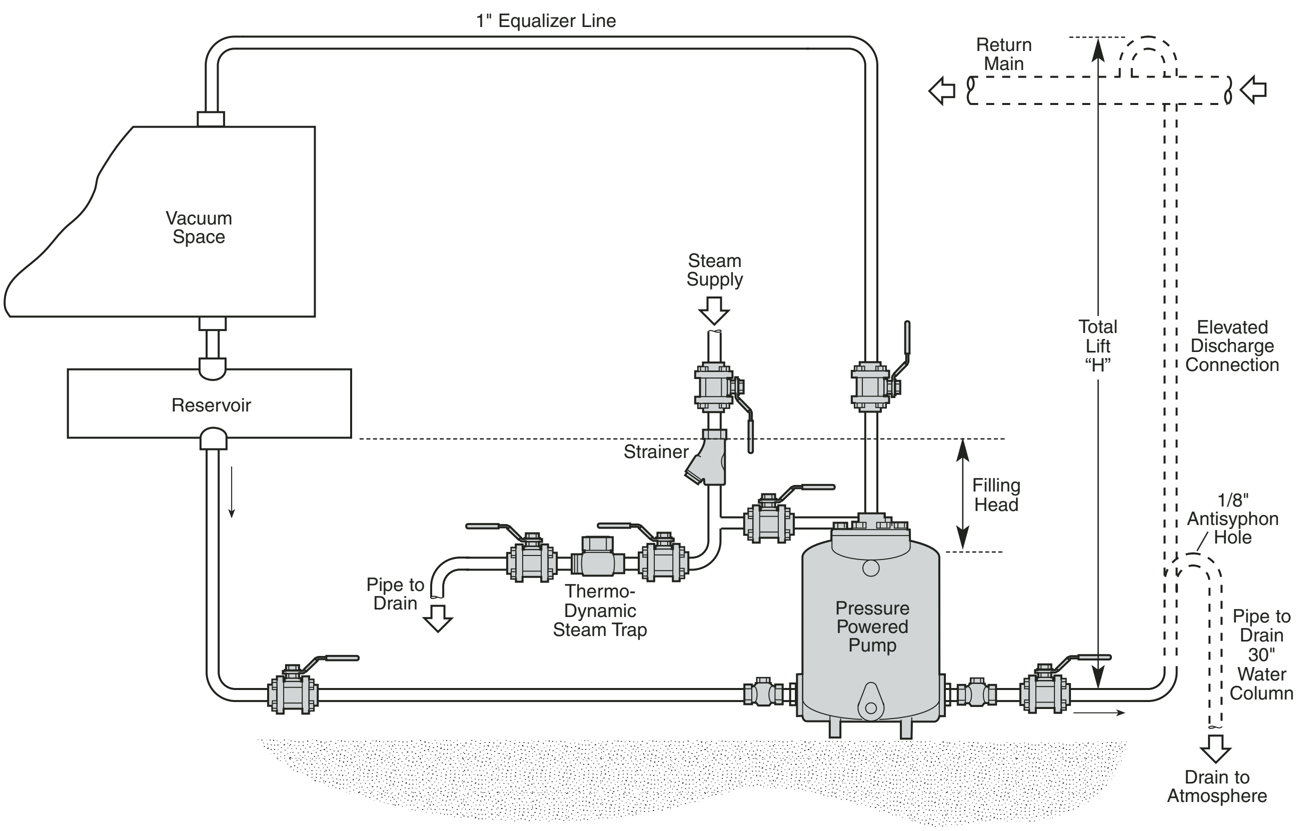
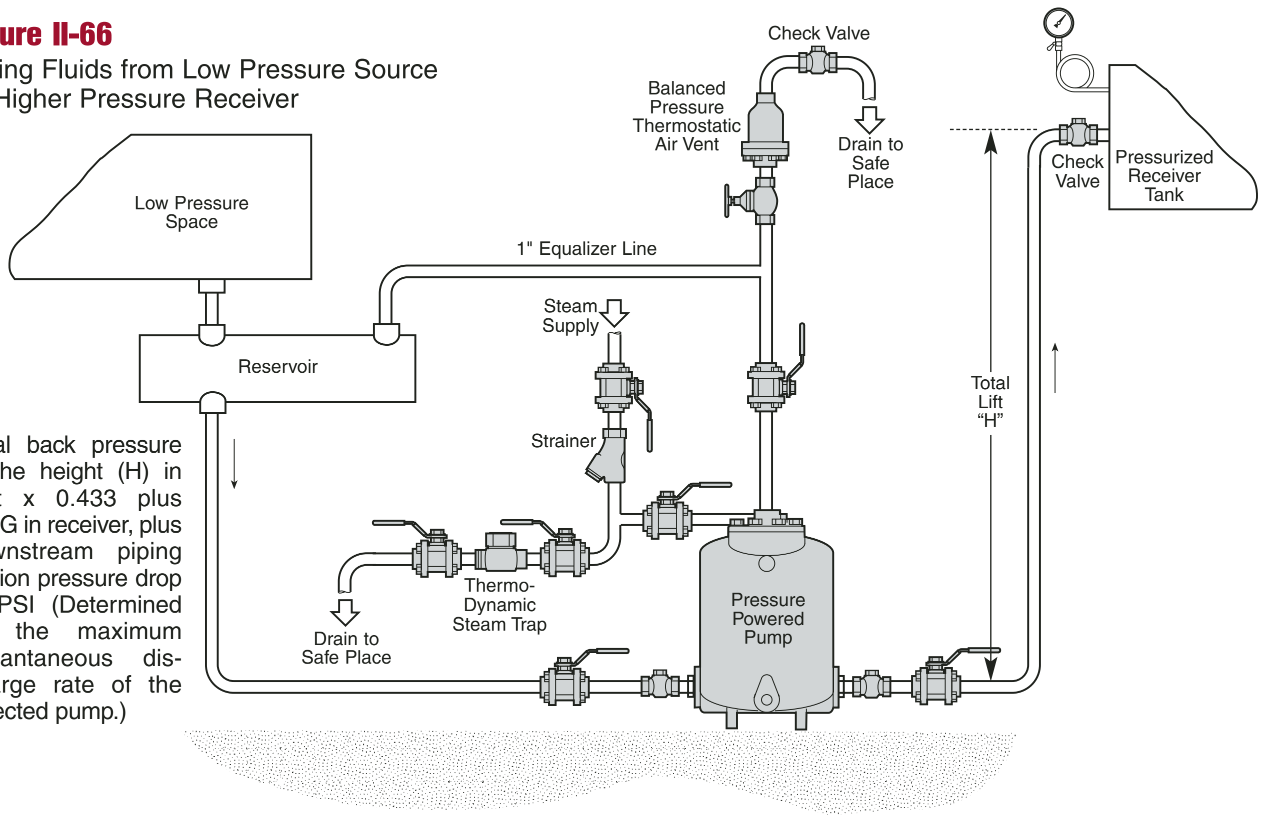


Figure II-66

Lifting Fluids from Low Pressure Source to Higher Pressure Receiver



Total back pressure is the height (H) in feet x 0.433 plus PSIG in receiver, plus downstream piping friction pressure drop in PSI (Determined by the maximum instantaneous discharge rate of the selected pump.)

Figure II-67

Draining Equipment with Condensate Outlet Near Floor Level using a Pump/Trap Combination in a Pit

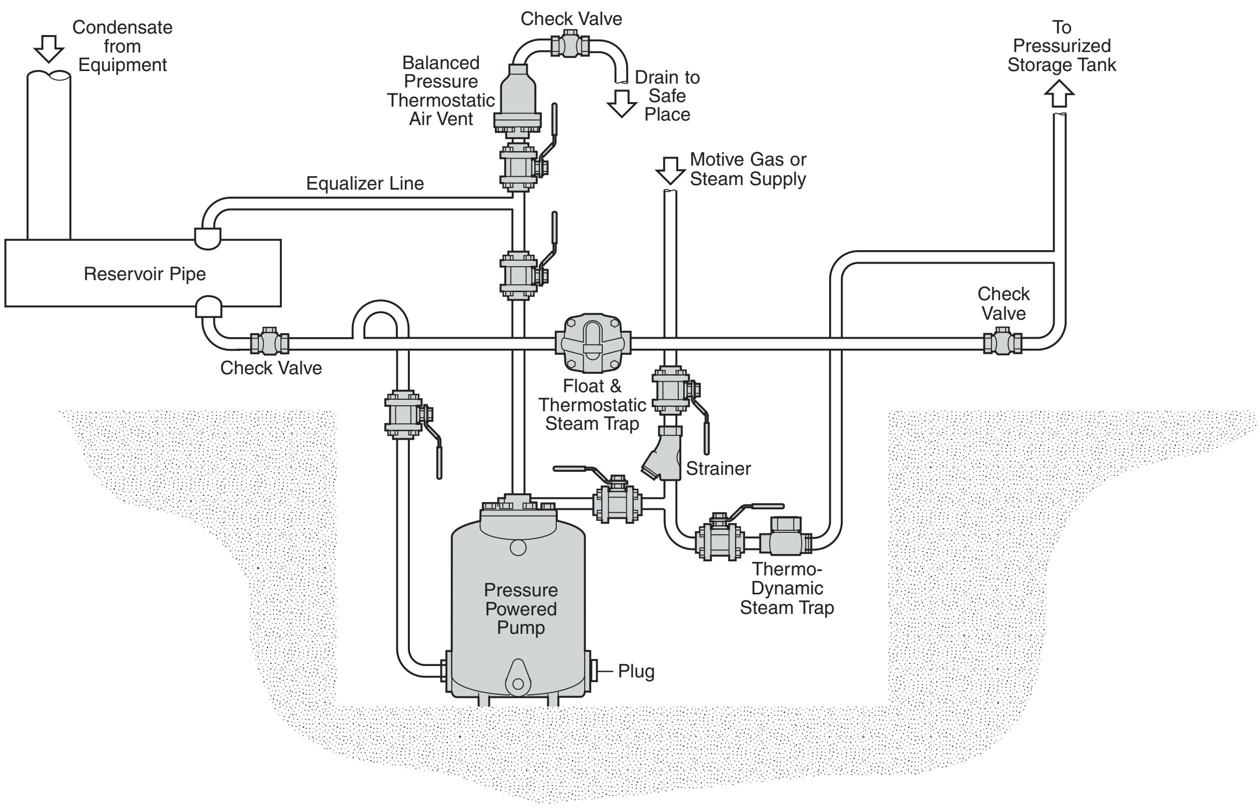


Figure II-68

Installation of Pump/Trap Combination when Vertical Space is Limited

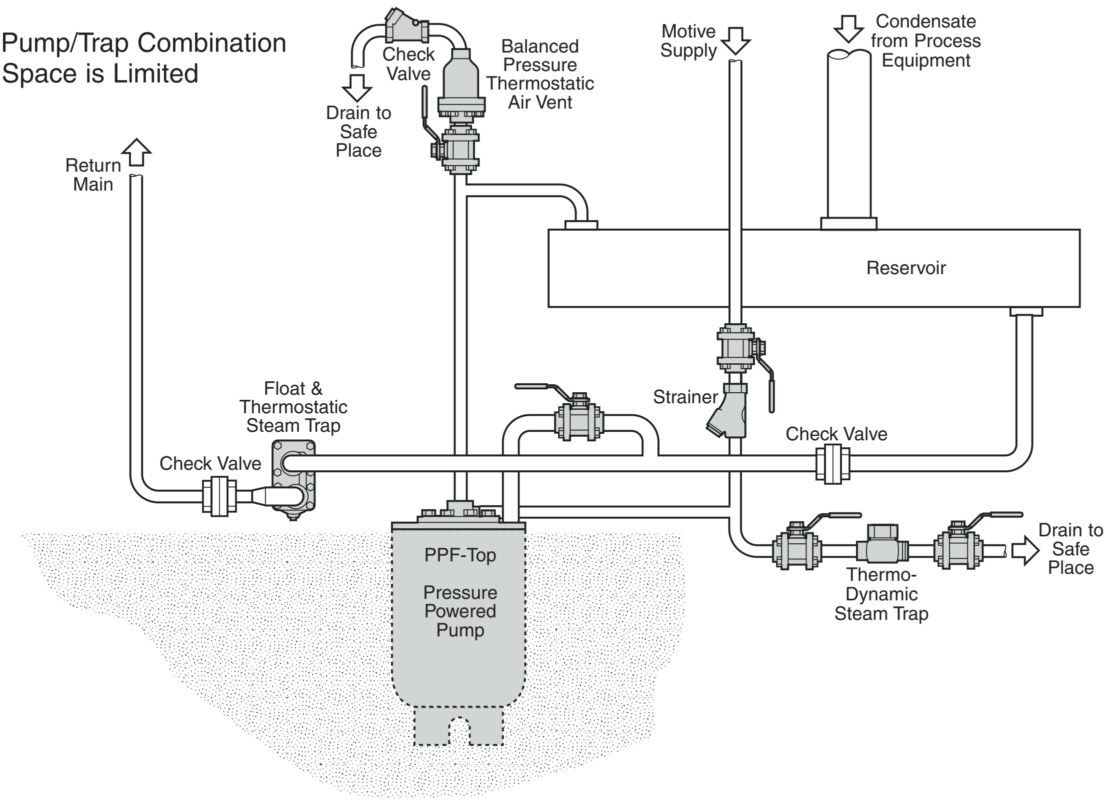


Figure II-69

Multiple Pressure Powered Pump Hookups for Staged Operation and Standby Duty

When multiple pumps in parallel are used to meet the load, Pump #1 fills first and Pump #2 operates at full load. Supply piping is arranged to reduce simultaneous discharge. Better staging occurs when Pump #2 is elevated 3" to 6" above the floor.

Intake pipe to Pump #3 is located above the Filling Head required by Pump #1 & #2 so it will function as a "Standby" and only operate on peak loads or in the event of primary pump failure.

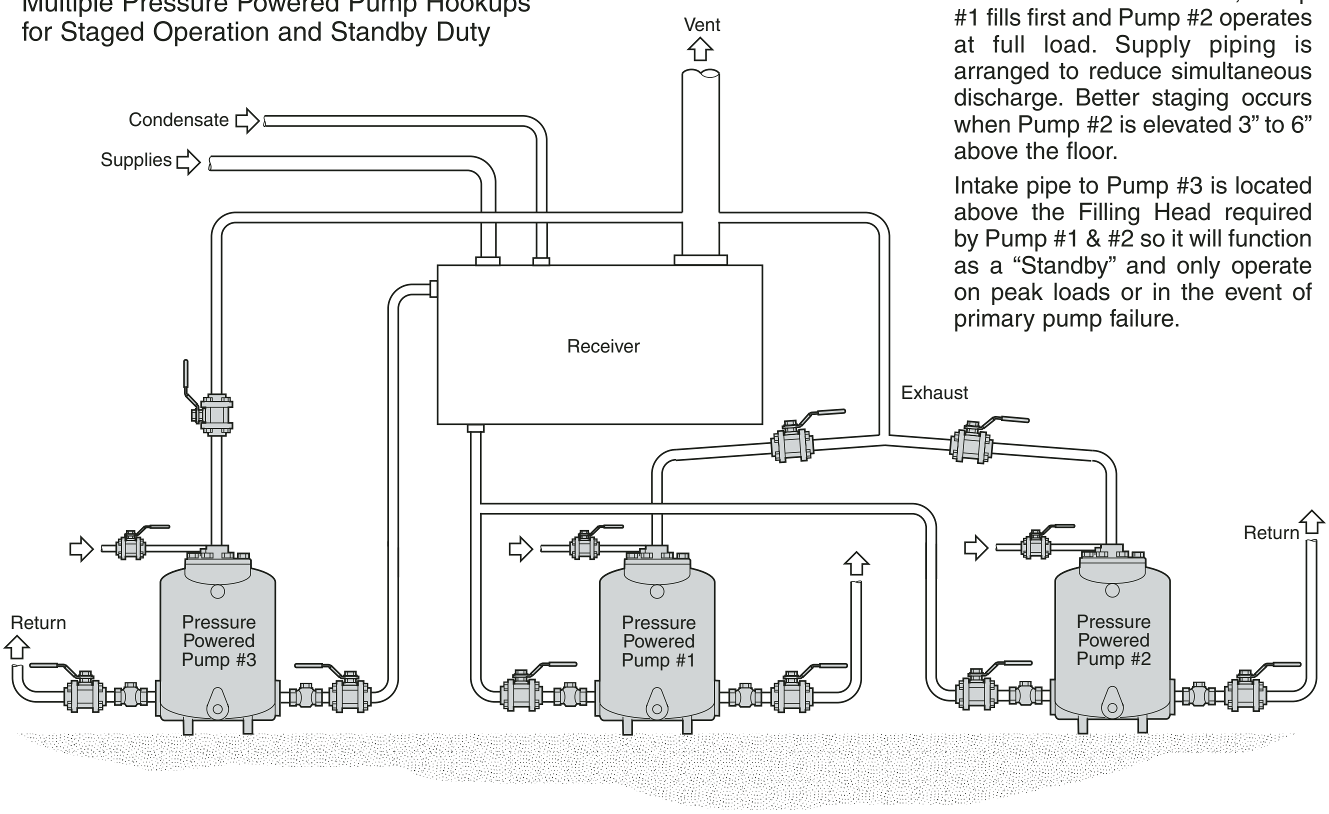


Figure II-70

Pressure Powered Pump Draining Water from Sump Pit

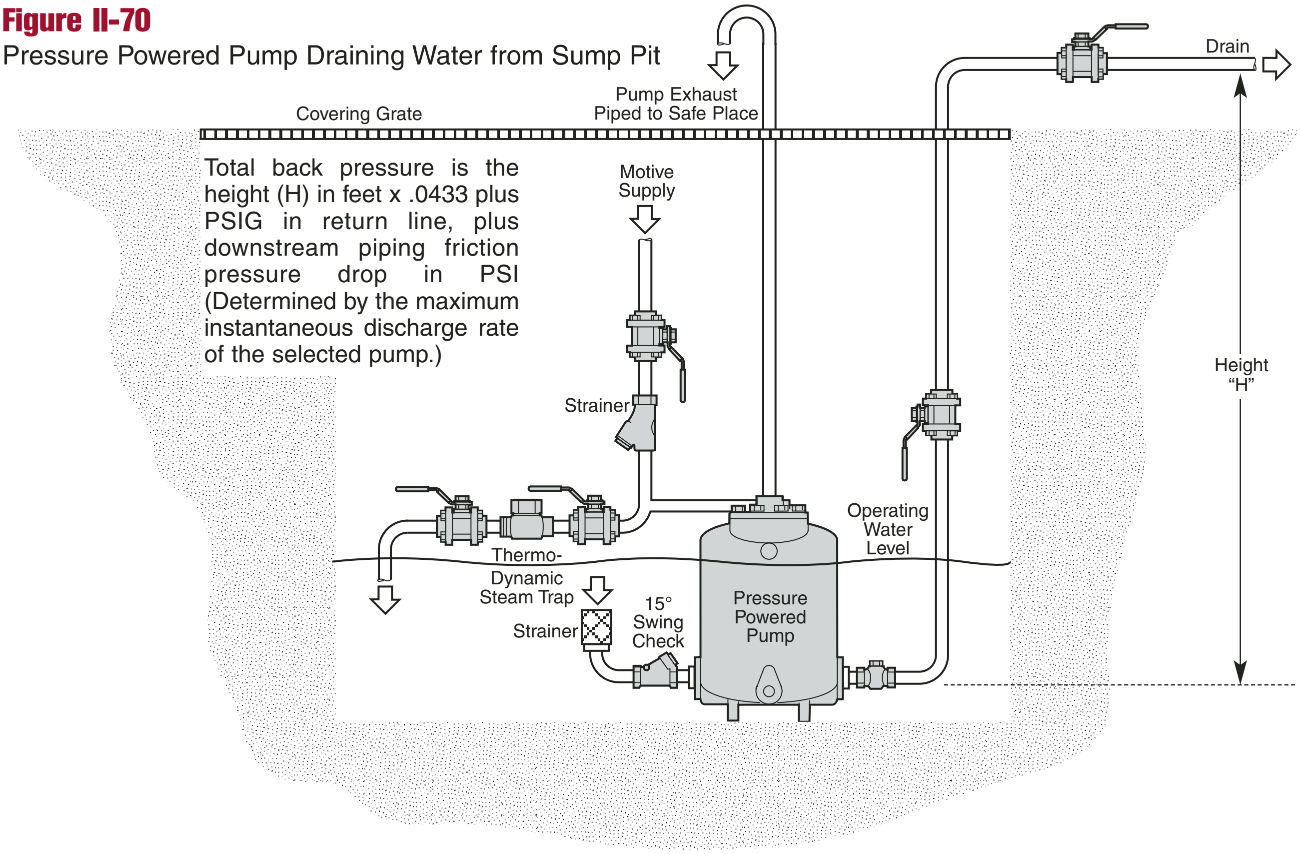


Figure II-71

Pressure Powered Pump Discharging to Long Delivery Line
(Air Eliminator needed above return main wherever elevation changes form a water seal.)

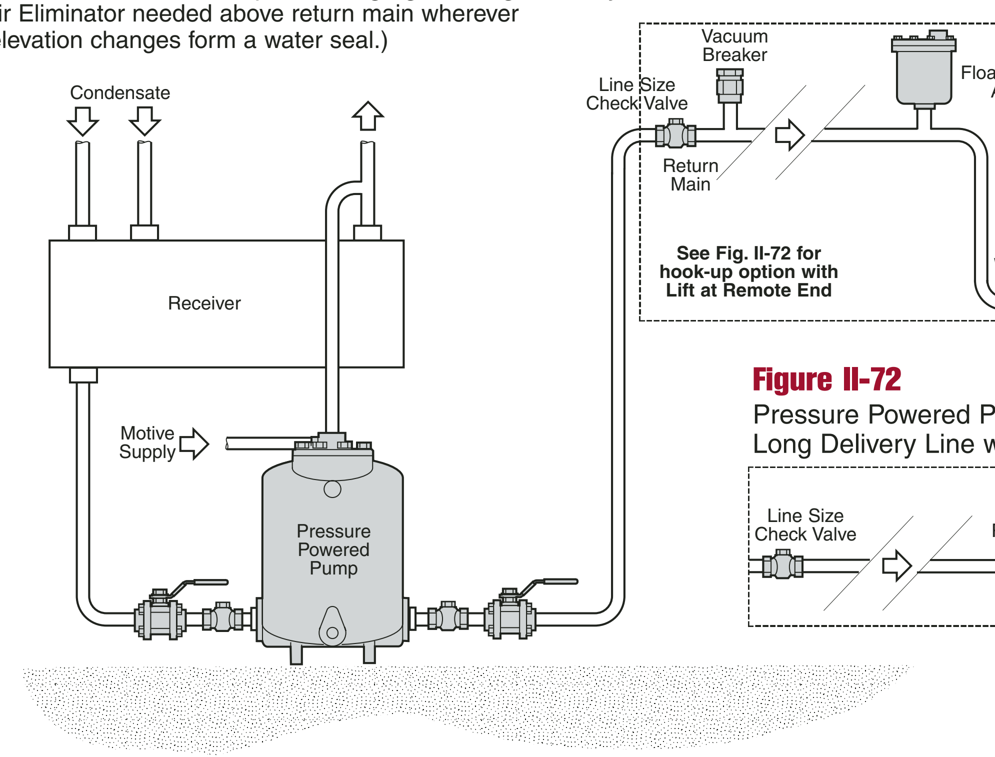


Figure II-72

Pressure Powered Pump Discharging to Long Delivery Line with Lift at Remote End

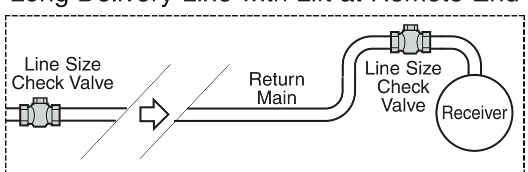
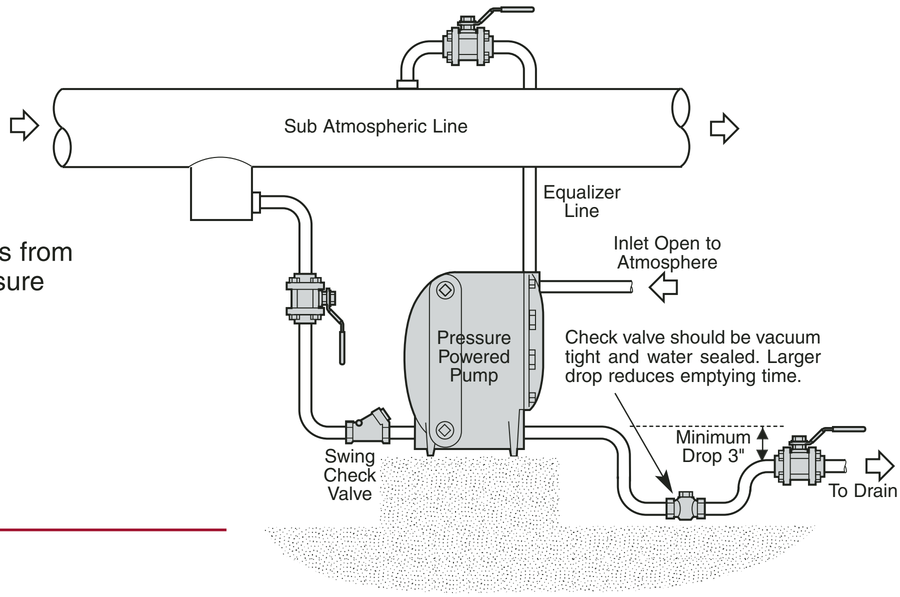


Figure II-73
Draining Small Condensate Loads from Vacuum using Atmospheric Pressure



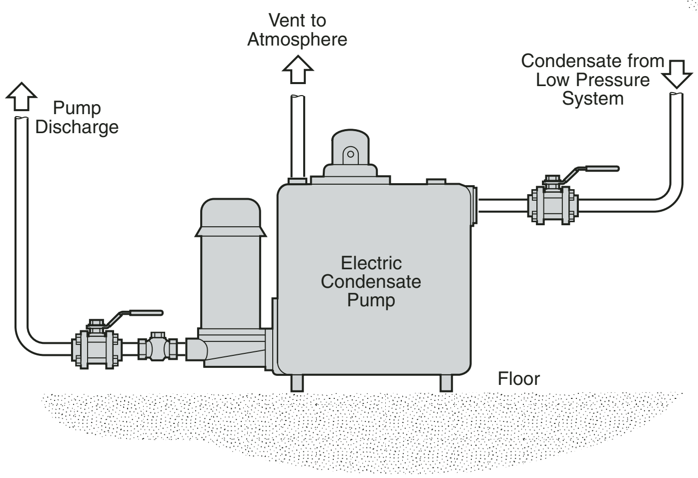
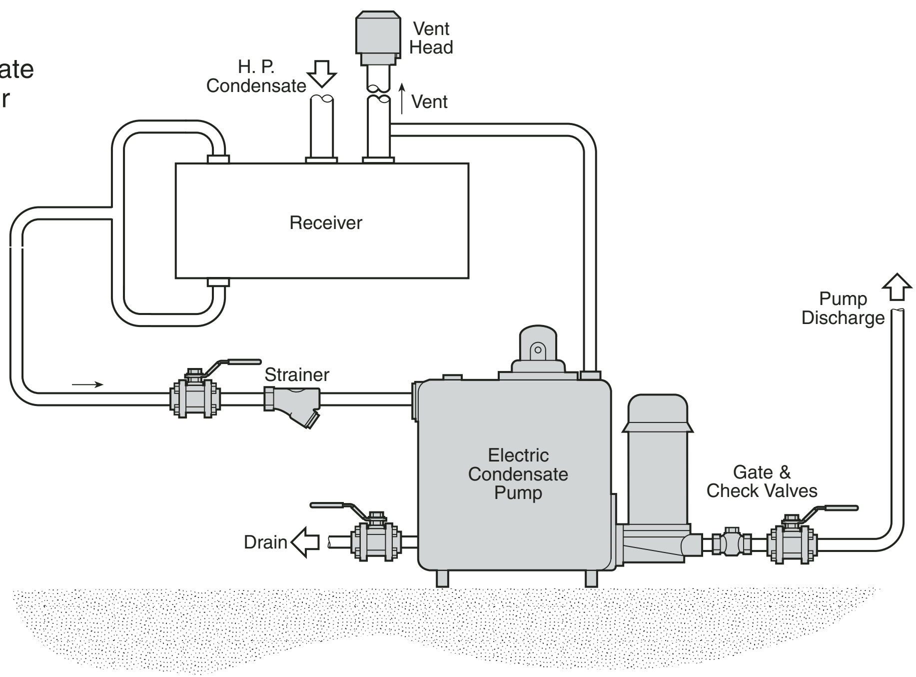


Figure II-74
Typical Electric Pump Hook-up for Subcooled Condensate

Figure II-75
Electric Pump Lifting Condensate from Vented Receiver to Higher Pressure or Elevation



Condensate from high pressure loads releases steam by flashing as it passes to the lower pressures, downstream of the high pressure traps. The mixture of steam and condensate is readily separated in Flash Vessels of appropriate dimensions and proportions. A supply of Low Pressure steam then becomes available for use on any application which can accept steam at this low pressure, or the separated steam may simply be taken into the LP steam mains, where it is supplemented through pressure reducing valves, for general plant use.

Where the supply of flash steam may at times exceed the demand from the LP system, the surplus flash steam can be discharged through a back pressure control valve. This is set at a few psi above the normal LP steam pressure, but below the setting of the LP safety valve. See Figure II-77.

The condensate leaving the flash recovery vessel is at low pressure. Usually it is handled by a float-thermostatic steam trap and is delivered to the receiver of a condensate pump for return to the boiler house. Any residual flash steam from the low pressure condensate is vented from the pump receiver. (Figure II-78.)

In some cases, pressures are sufficiently high that the flash can be taken off at an intermediate pressure and the condensate leaving the flash vessel still contains a useful amount of sensible heat. It can then be taken to a second flash vessel working at low pressure, so that the maximum heat recovery is effected. The use of two flash vessels in series, or “cascade”, means that these vessels may be installed generally as Figure II-76 and II-77.

Alternatively, it may be desirable to use the recovered flash steam at a low pressure, below that in the condensate return line or perhaps the de-aerator tank. The arrangement adapted may then be either as Figure II-78 or as Figure II-79. This latter system uses a steam powered pump, with the bottom of the flash recovery vessel serving as the pump receiver. Power steam used by the pump is vented to the LP steam line, so that pumping is achieved at virtually zero cost and the use of unsightly or wasteful vents is avoided.

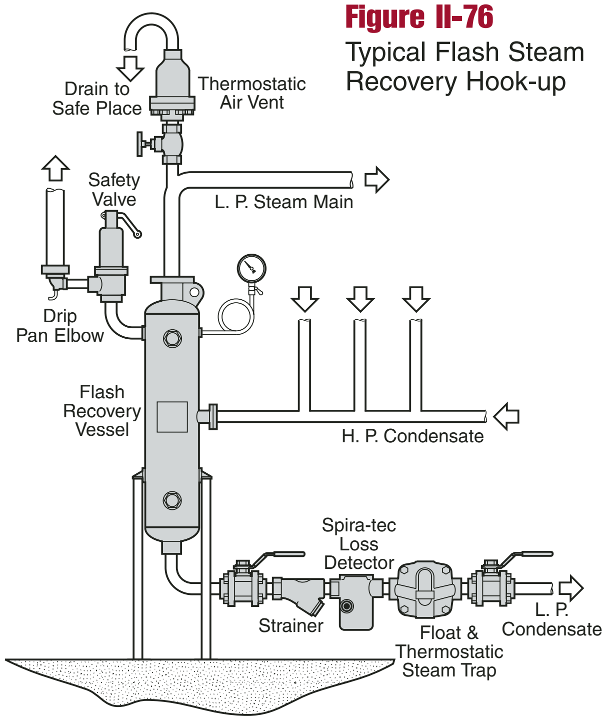


Figure II-76
Typical Flash Steam Recovery Hook-up

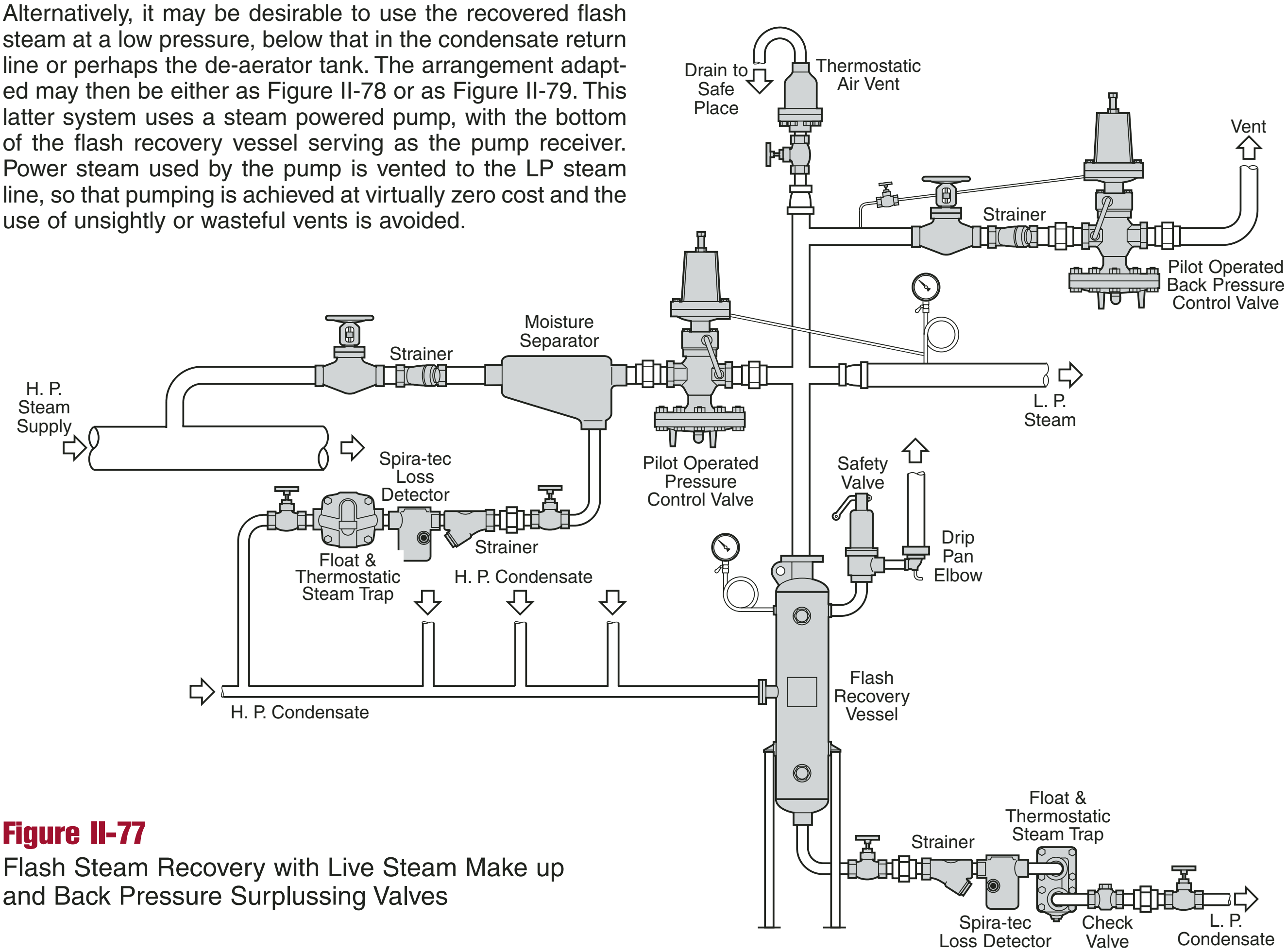
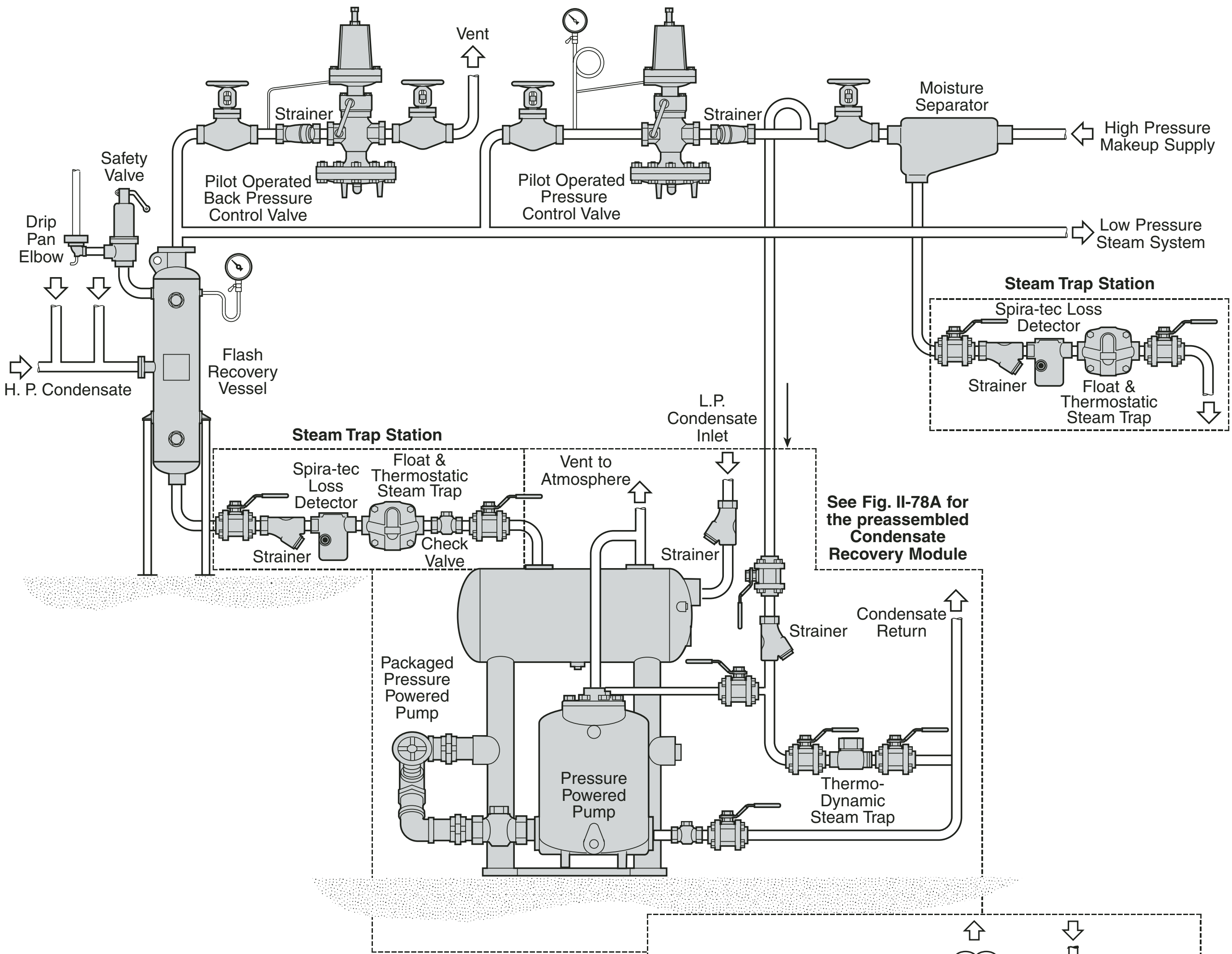


Figure II-77
Flash Steam Recovery with Live Steam Make up and Back Pressure Surplussing Valves

Figure II-78

Flash Steam Recovery at Pressure above Atmospheric with L.P. Condensate Returned by Packaged Pressure Powered Pump Unit



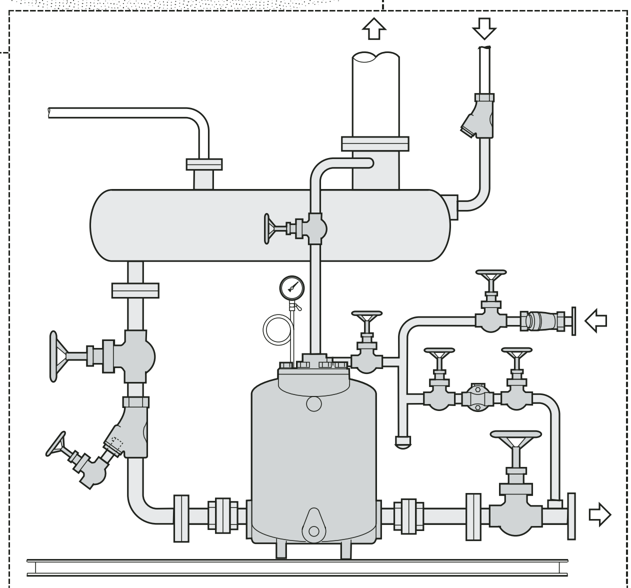


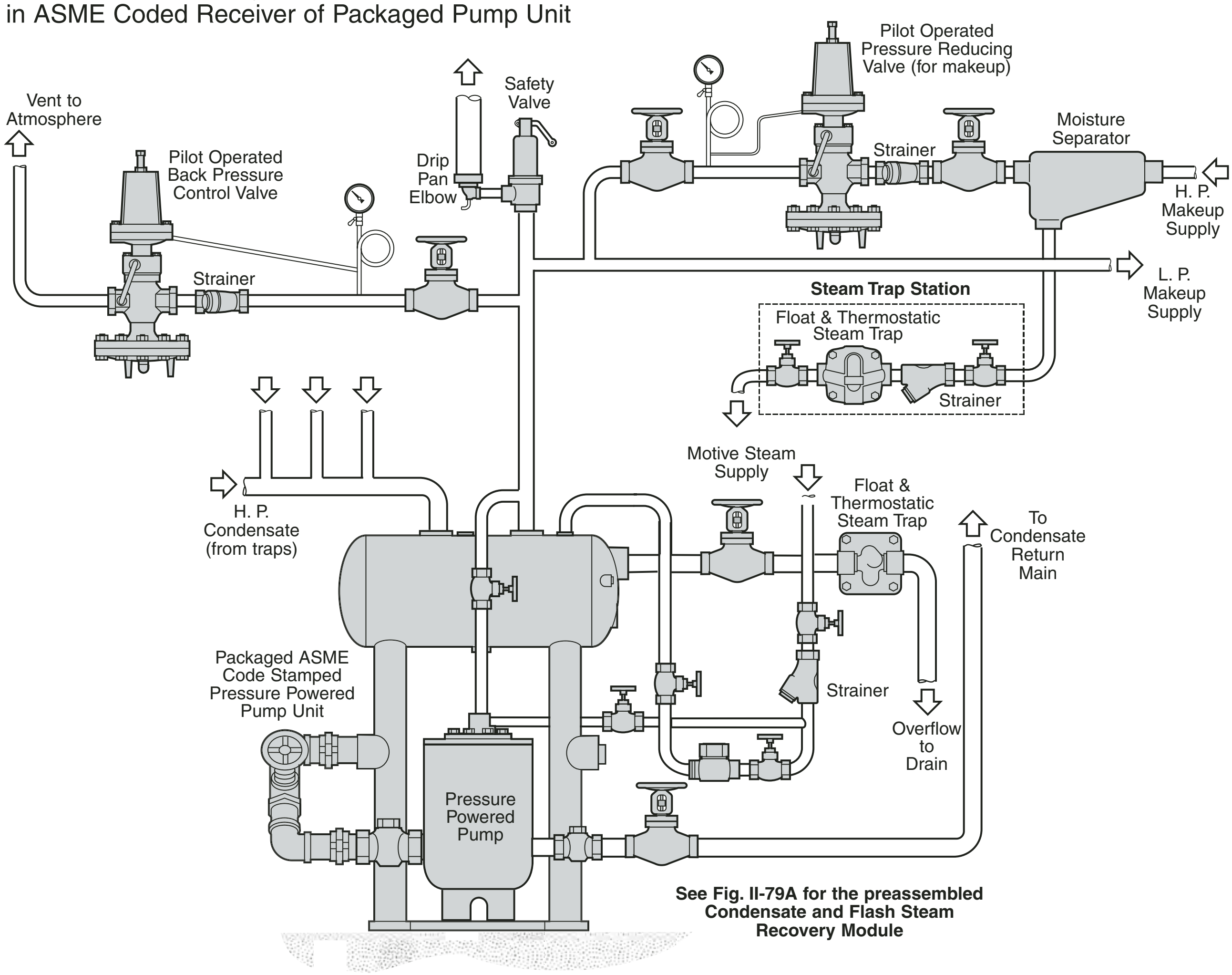
Figure II-78A

Condensate Recovery Module

A preassembled modular pumping system provides a sole source solution for condensate recovery applications.

Figure II-79

Flash Steam Recovery at Pressure Above or Below Atmospheric in ASME Coded Receiver of Packaged Pump Unit



See Fig. II-79A for the preassembled Condensate and Flash Steam Recovery Module

Figure II-79A

Condensate and Flash Steam Recovery Module

A preassembled modular pumping system will recover condensate and direct flash steam to a low pressure user.

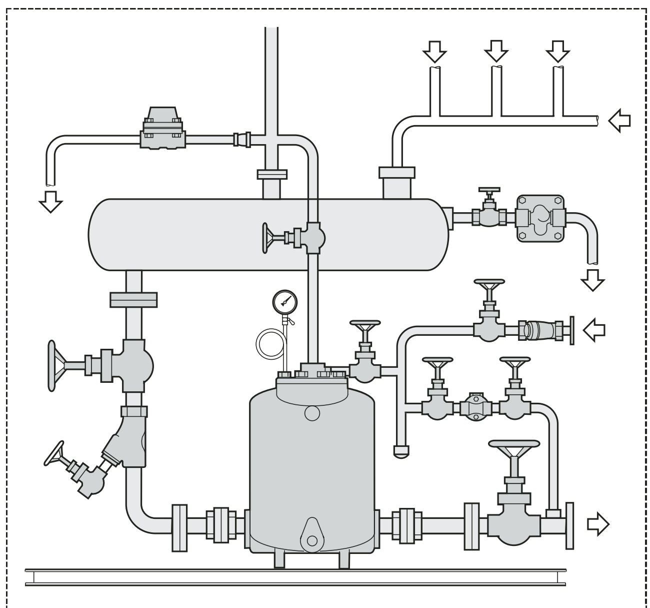
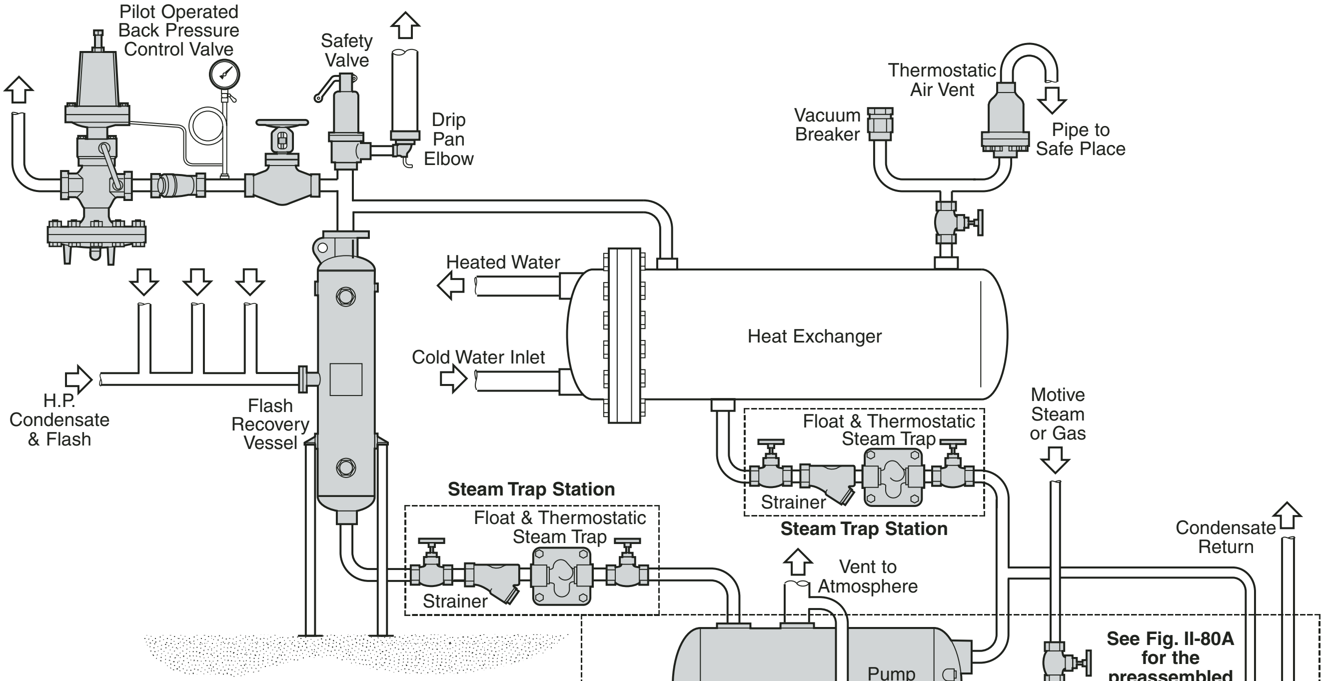
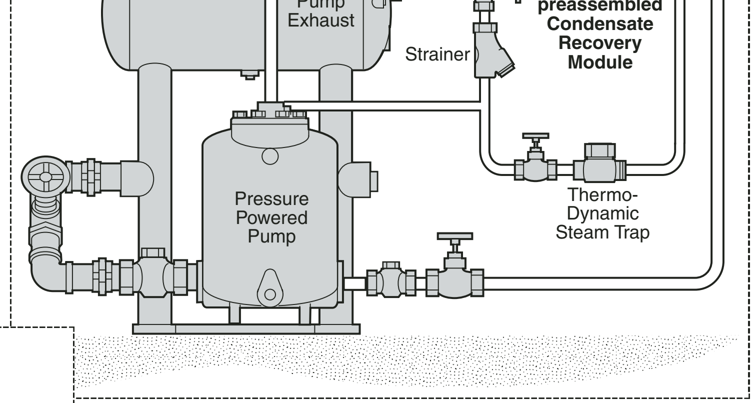


Figure II-80

Heating Water using Recovered Flash Steam with Packaged Pump Unit Also Handling Other Condensate



Condensate from high pressure steam sources is collected in a flash steam recovery vessel. Flash steam is separated from the condensate which drains through an F & T trap set to the vented receiver of a condensate pump. The flash steam is condensed in a heat exchanger, giving up its heat content to the water which is to be heated (or preheated). Condensate from the exchanger also drains through an F & T trap set to the pump receiver. Non-condensibles are discharged at the receiver vent.



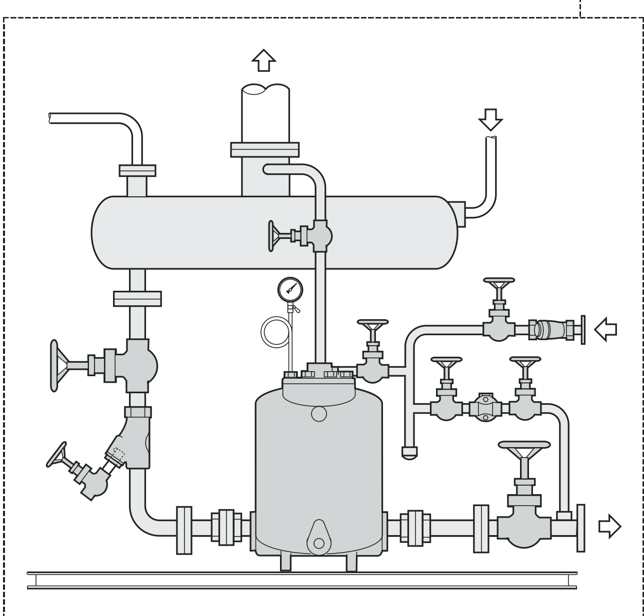


Figure II-80A

Condensate Recovery Module

A preassembled modular pumping system can be used to recover and reuse the condensate.

Figure II-81

Heating Water using Flash Steam Recovered in ASME Coded Receiver of Packaged Pressure Powered Pump Unit

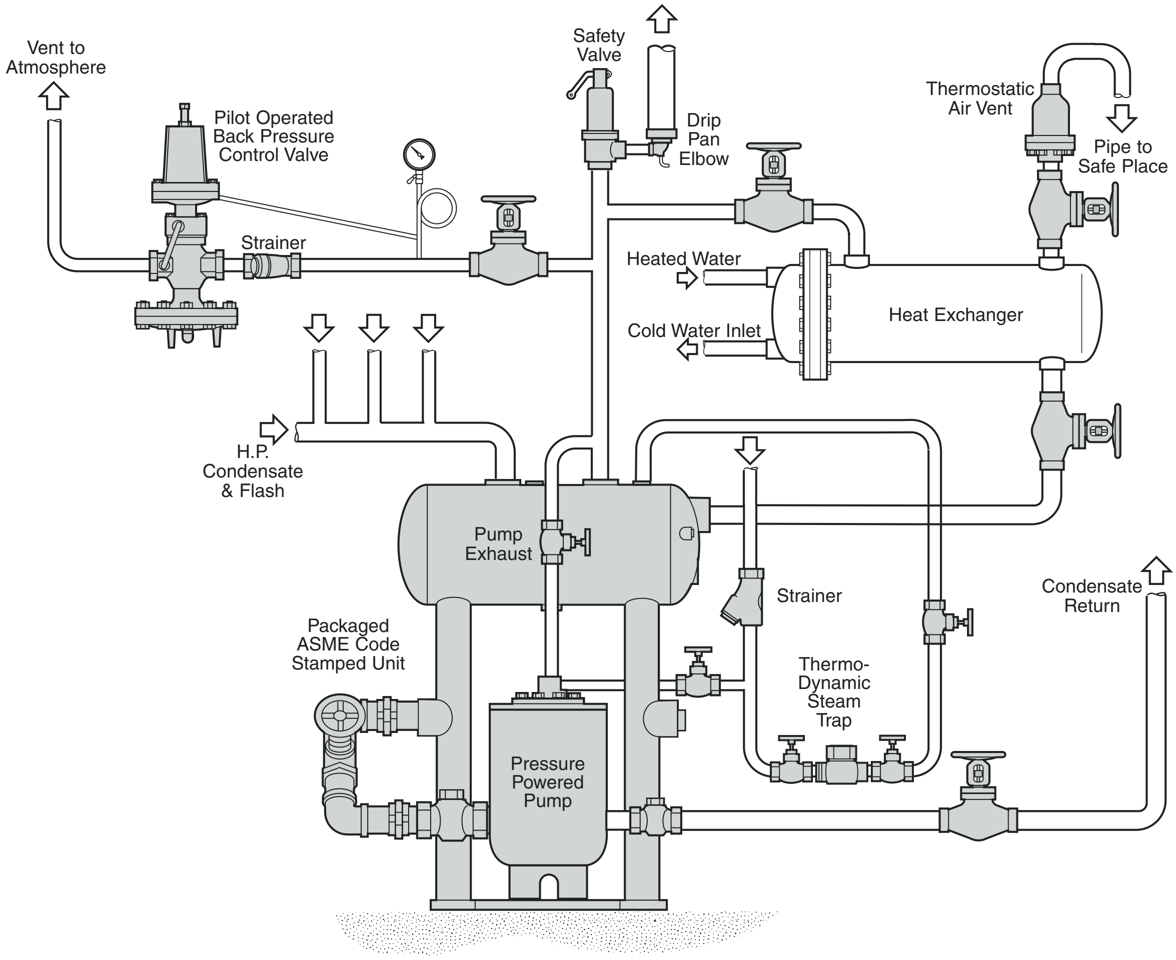
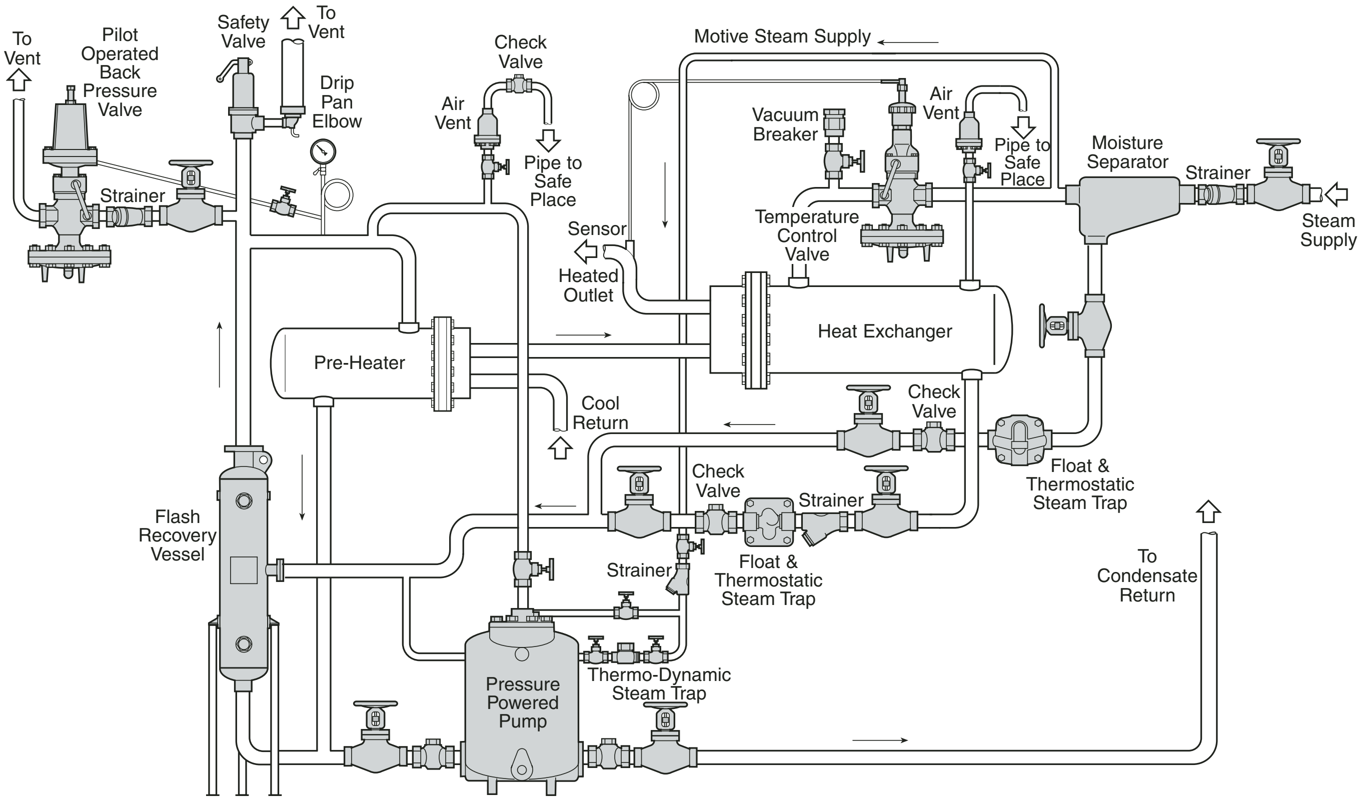


Figure II-82

Recovery of Flash Steam and Pump Power Steam on Preheater (Steam in the Shell)



Condensate from the main heat exchanger flows to a flash Steam Recovery Vessel. The flash steam is separated and led to the Preheater where it is condensed as it preheats the incoming cool water or other liquid. Any incondensibles are discharged to atmosphere through the thermostatic air vent.

Residual condensate from the flash vessel, with that from the preheater, falls to the inlet of the Pressure Powered Pump. Pump exhaust steam is taken to the Flash Steam line and its heat content recovered in the preheater.

A Packaged Pump Unit with ASME coded receiver can be used in place of the component flash vessels and Pressure Powered Pump.

Figure II-83

Recovery of Flash Steam and Pump Power Steam in Preheater (Steam in Tubes)

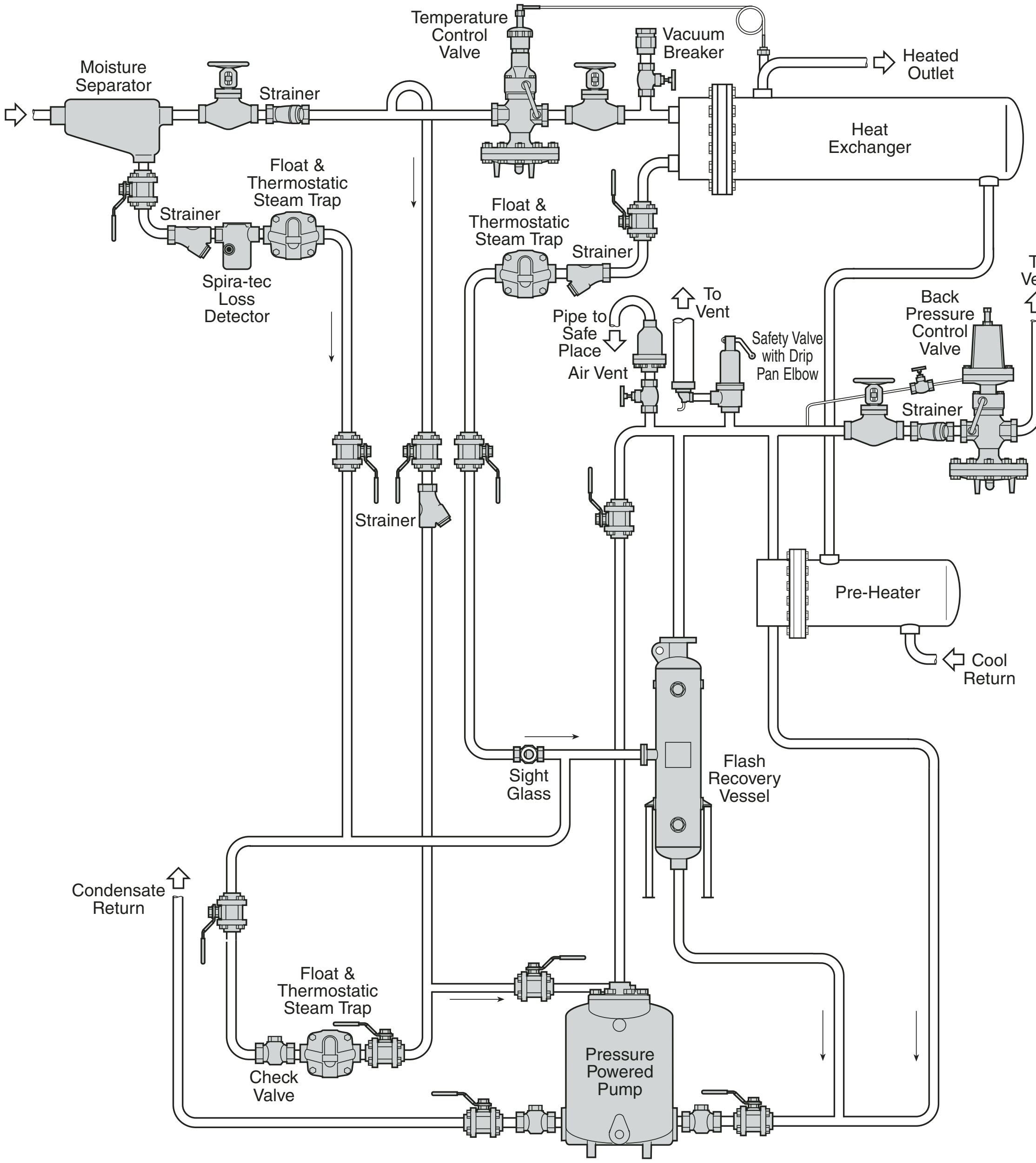
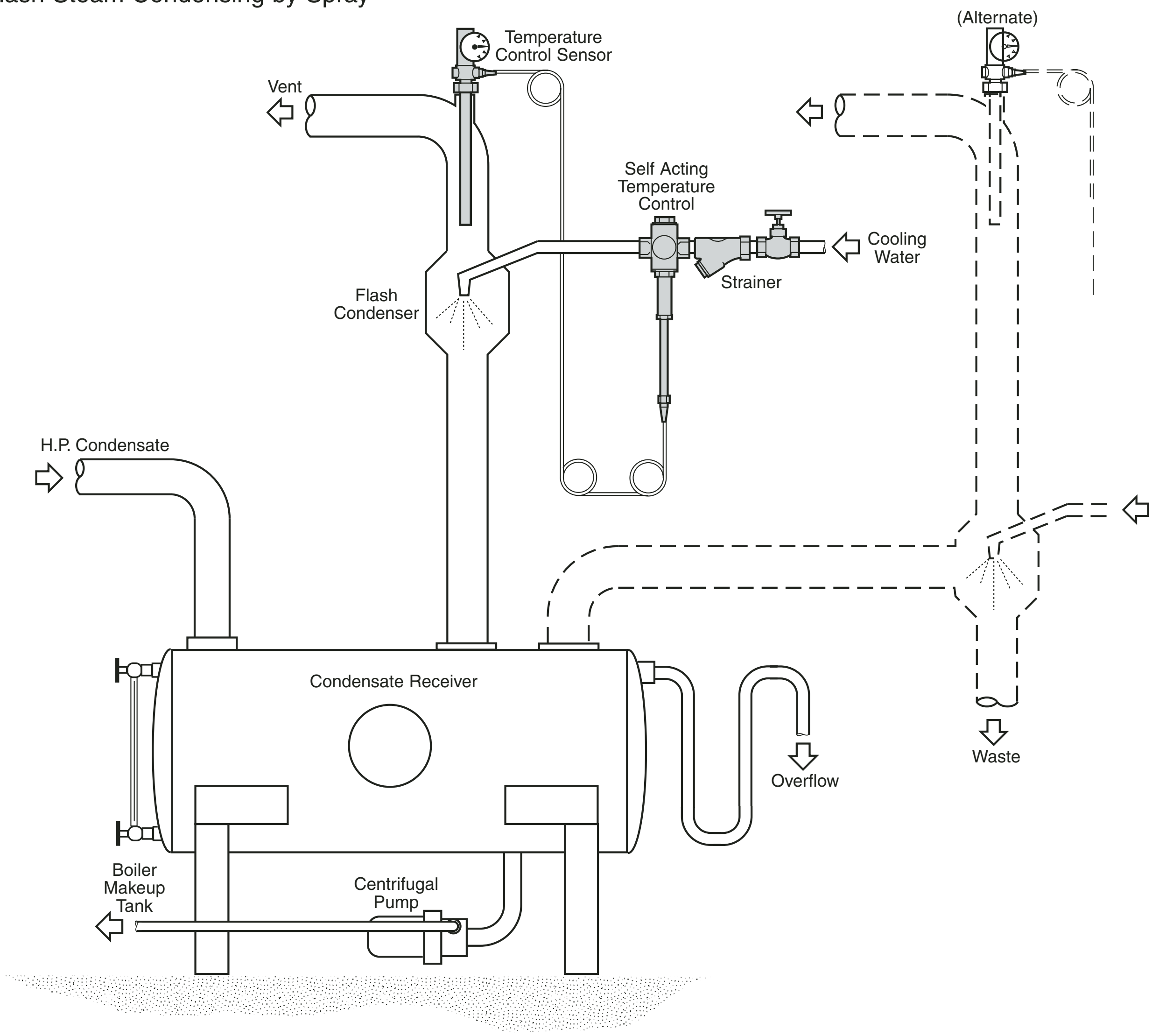


Figure II-84

Flash Steam Condensing by Spray

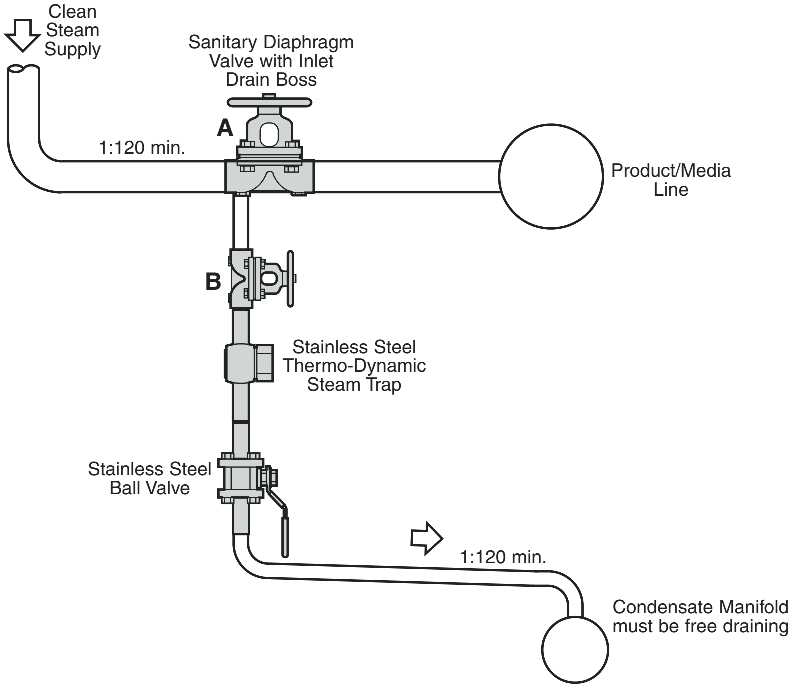


Residual flash steam for which no use can found often causes a nuisance if vented to atmosphere, and of course carries its valuable heat content with it. This steam may be condensed by spraying in cold water, in a light gauge but corrosion resistant chamber fitted to the receiver tank vent. If boiler feed quality water is used, the warmed water and condensed flash steam is added to the condensate in the receiver and reused. Condensing water which is not of feed water quality is kept separate from the condensate in the receiver as shown dotted.

A self-acting, normally closed temperature control with sensor in the vent line can control the coolant flow. This minimizes water usage, and where condensed flash steam is returned, avoids overcooling of the water in the receiver.

Figure II-85

Clean Steam Drip Station

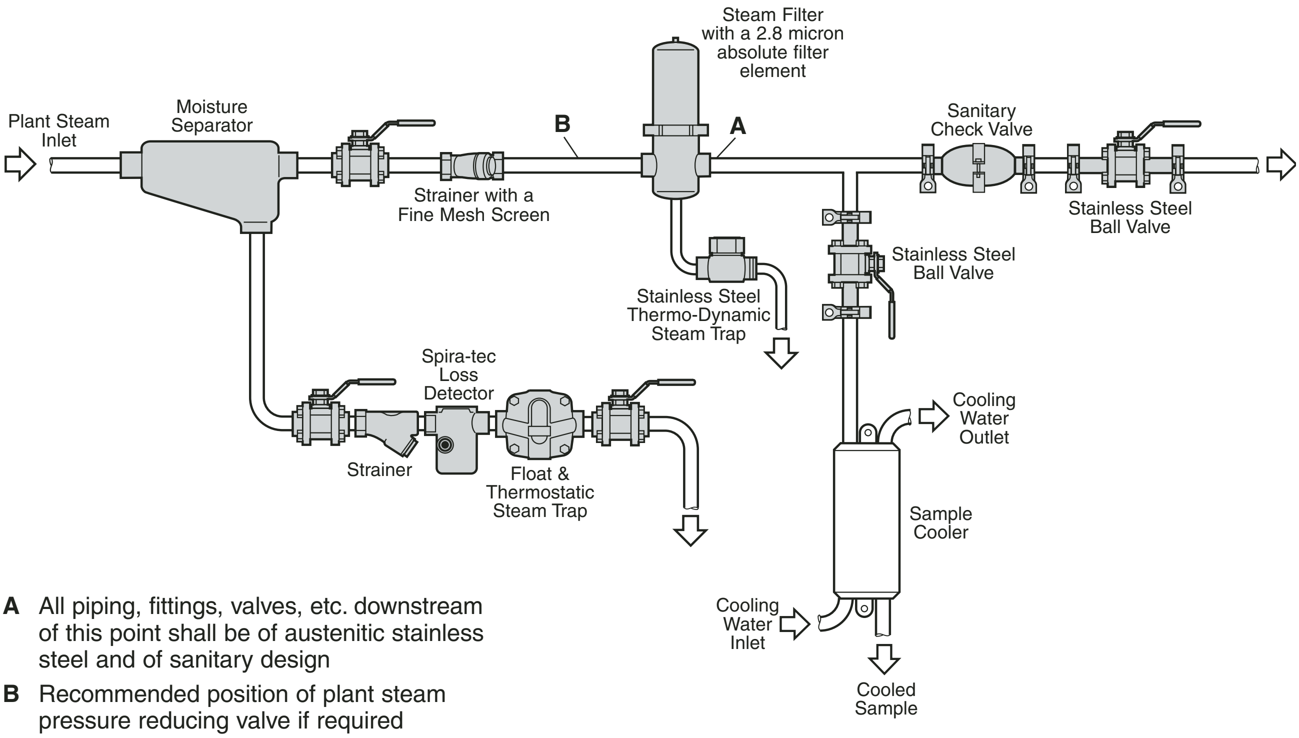


Install valve "**A**" close to product/media line.

Close-couple valves "**A**" and "**B**"

Figure II-86

Culinary/Filtered Steam Station



A All piping, fittings, valves, etc. downstream of this point shall be of austenitic stainless steel and of sanitary design

B Recommended position of plant steam pressure reducing valve if required

Pressure gauges, fittings, valves, etc., are not shown for clarity.

Figure II-87 Tank Sterilization

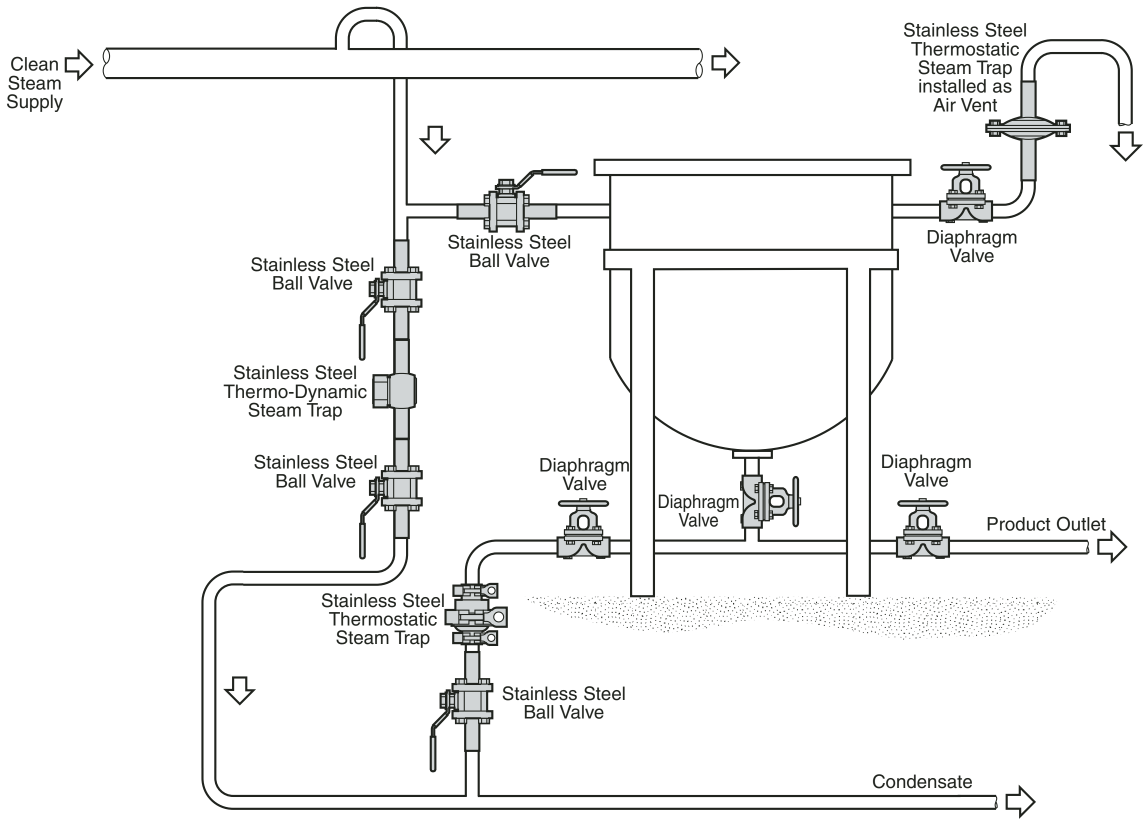
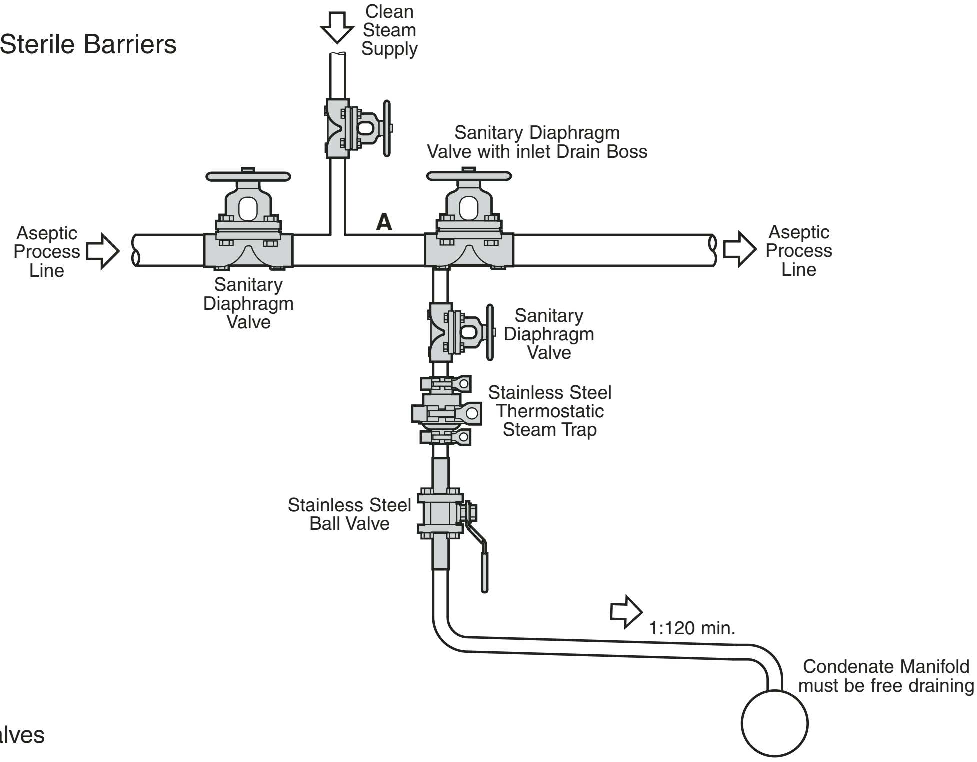


Figure II-88
Block and Bleed Sterile Barriers



A Close-couple valves

HOOK-UP DIAGRAMS

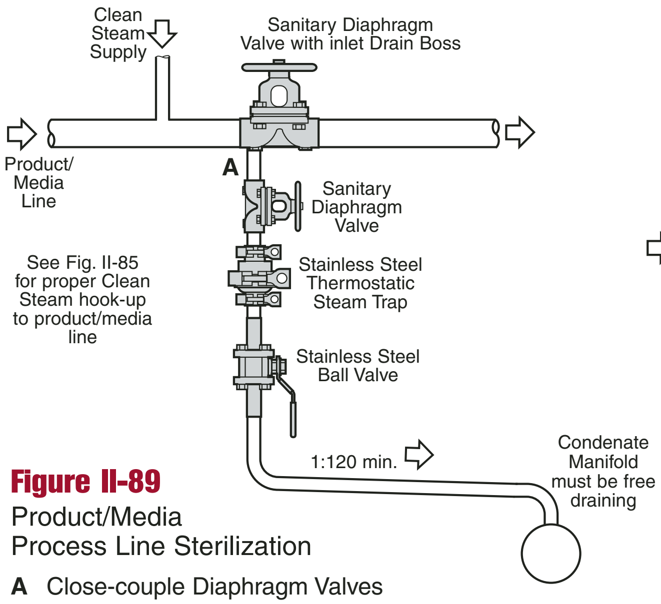


Figure II-89

Product/Media
Process Line Sterilization

A Close-couple Diaphragm Valves

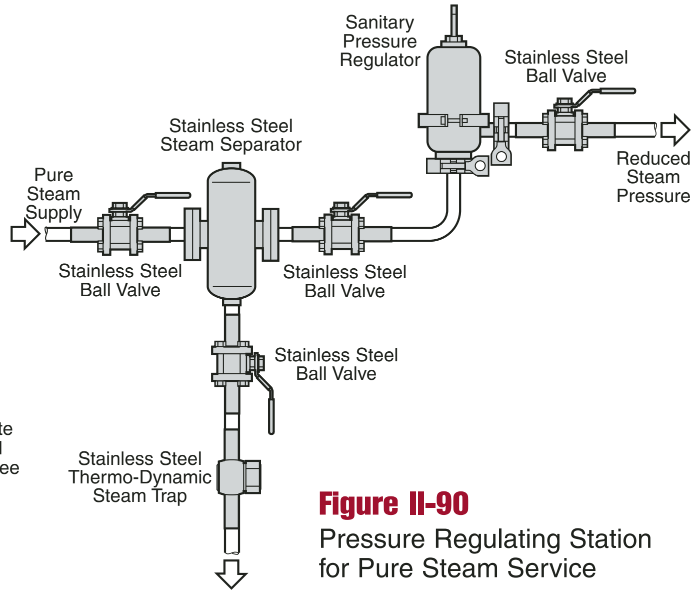
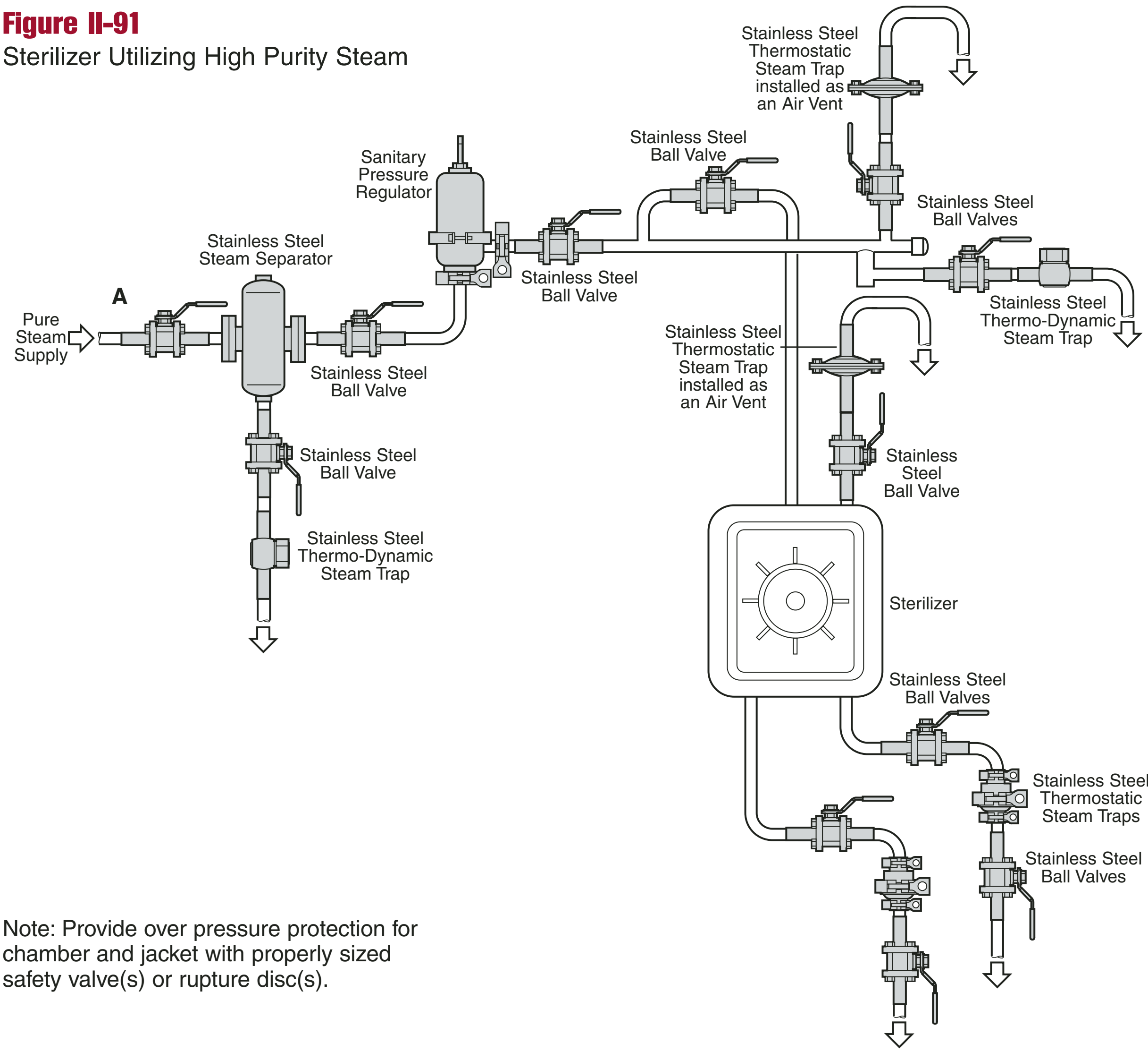


Figure II-90

Pressure Regulating Station
for Pure Steam Service

Figure II-91

Sterilizer Utilizing High Purity Steam



Note: Provide over pressure protection for chamber and jacket with properly sized safety valve(s) or rupture disc(s).

Figure II-92

Spiraflo Saturated Steam (Density Compensated) Metering System

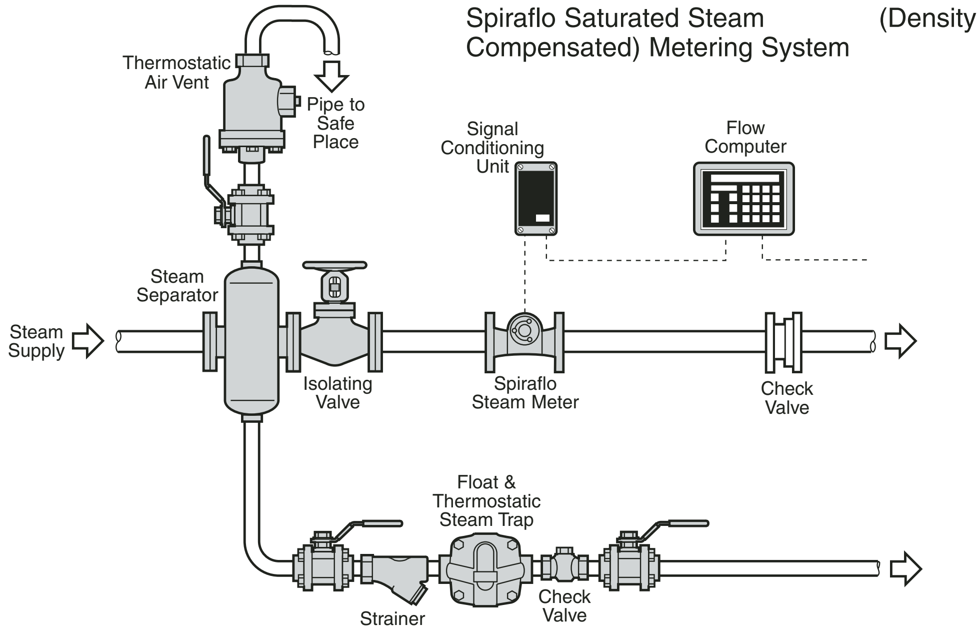
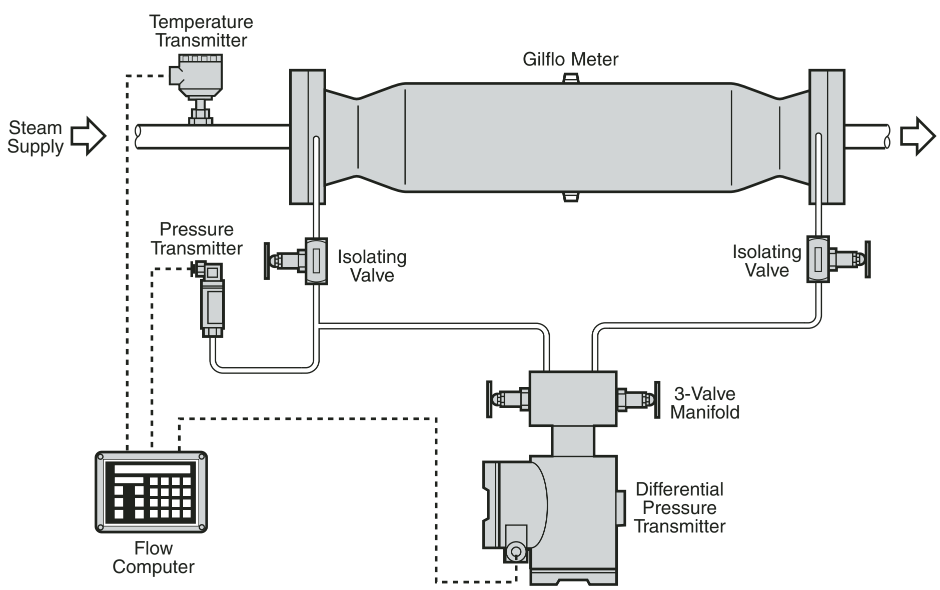


Figure II-93

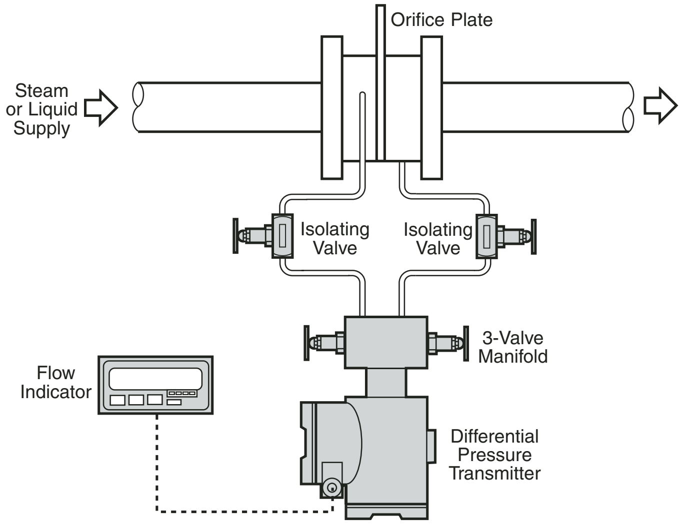
Typical Superheated Steam (Density Compensated) Metering System



Note: The same configuration is suitable for the Standard Range Gilflo, Gilflo ILVA and Orifice Plate Systems.

Figure II-94

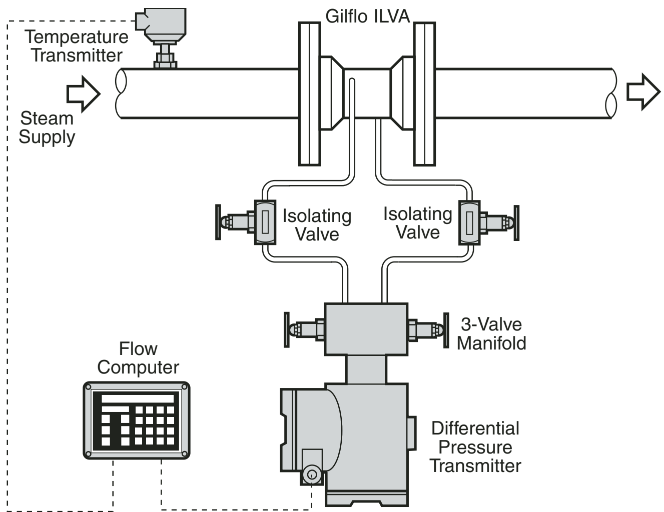
Typical Saturated Steam or Liquid Metering System (No Density Compensation)



Note: The same configuration is suitable for the Gilflo, Standard Range Gilflo, and Gilflo ILVA Systems.

Figure II-95

Typical Saturated Steam (Density Compensated) Metering System

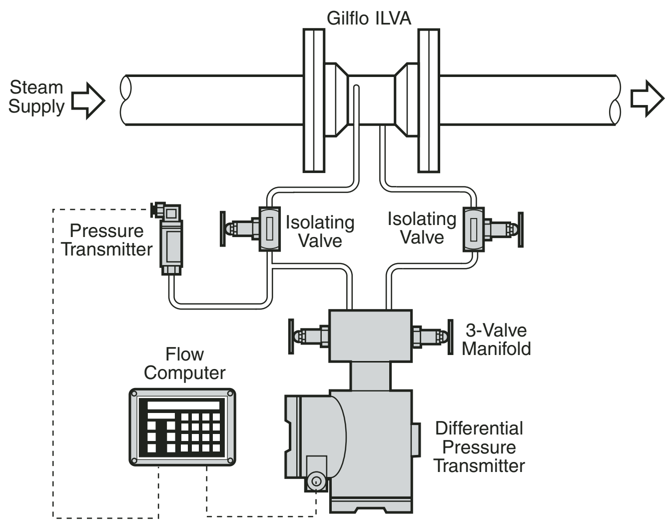


Note: The same configuration is suitable for the Gilflo, Standard Range Gilflo, and Orifice Plate Systems.

For Saturated Steam, Density Compensation is achieved by the flow computer accepting a signal from either a Temperature Transmitter (as shown here) or a Pressure Transmitter (see Fig. II-96)

Figure II-96

Typical Saturated Steam (Density Compensated) Metering System



Note: The same configuration is suitable for the Gilflo, Standard Range Gilflo, and Orifice Plate Systems.

For Saturated Steam, Density Compensation is achieved by the flow computer accepting a signal from either a Pressure Transmitter (as shown here) or a Temperature Transmitter (see Fig. II-95)

Figure II-97

Hand Operated Rotary Filter

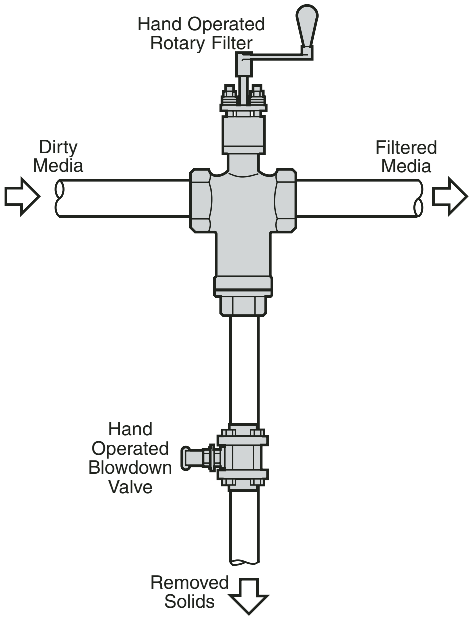
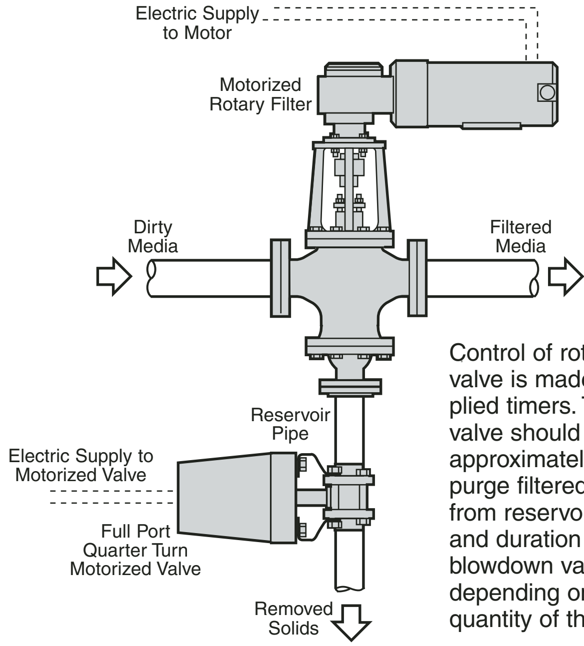


Figure II-98

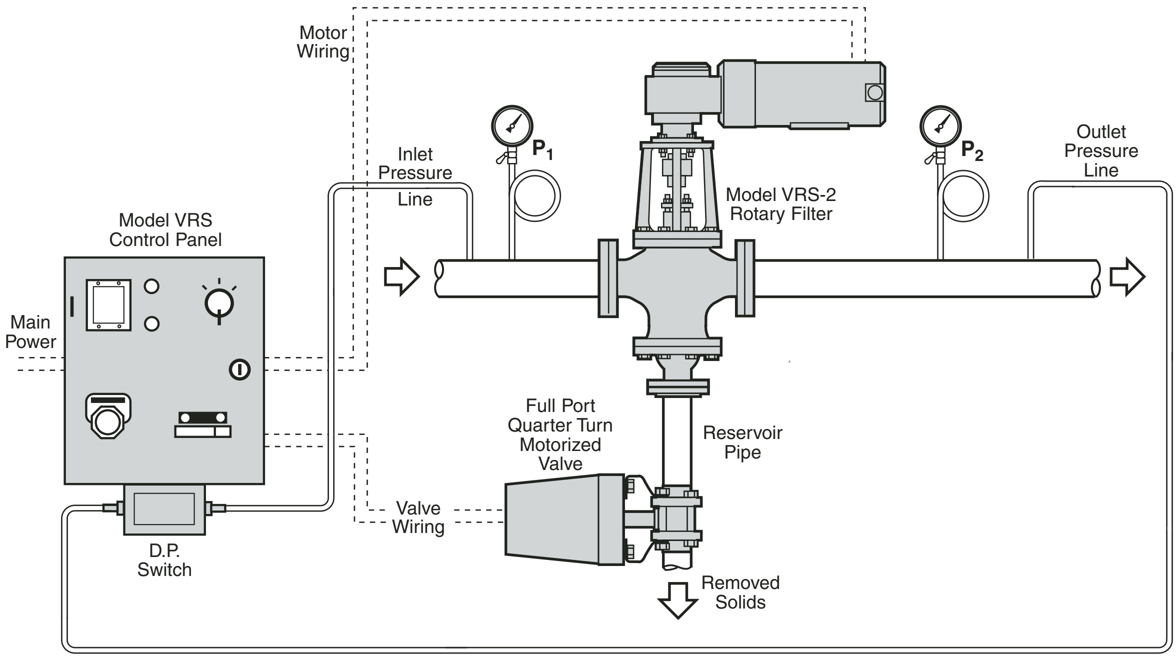
Motorized Rotary Filter with Single Blowdown Valve



Control of rotor and blowdown valve is made with user supplied timers. The blowdown valve should stay open for approximately 10 seconds to purge filtered dirt and debris from reservoir pipe. Interval and duration of the rotor and blowdown valve will vary depending on the nature and quantity of the dirt and debris.

Figure II-99

Control Panel Hook-up for One Valve Blowdown VRS-2 Rotary Filter System



The Control Panel with user adjustable timers controls interval and duration of the rotor and blowdown valve operation. The blowdown valve opens for approximately 10 seconds to purge the reservoir pipe of filtered dirt and debris. The Control Panel is shown with optional cycle counter and differential pressure switch which will activate rotor operation if excessive pressure drop occurs. This hook-up illustrates an automatic filtration system, providing continuous production flow with minimal product loss.

Figure II-100

Control Panel Hook-up and Operation of Two Valve Blowdown VRS-2 Rotary Filter System

Figure II-100A

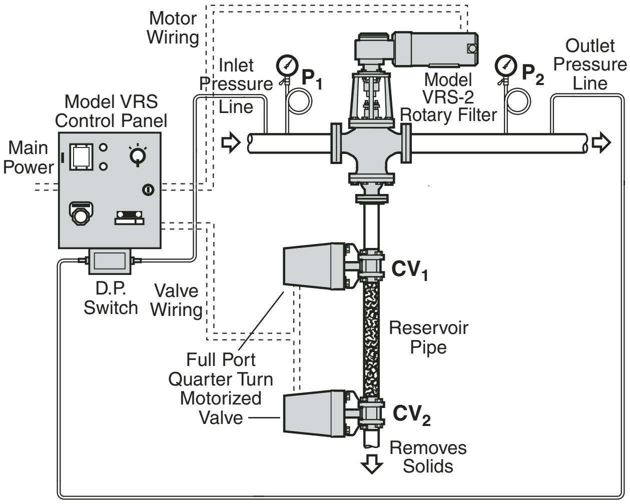
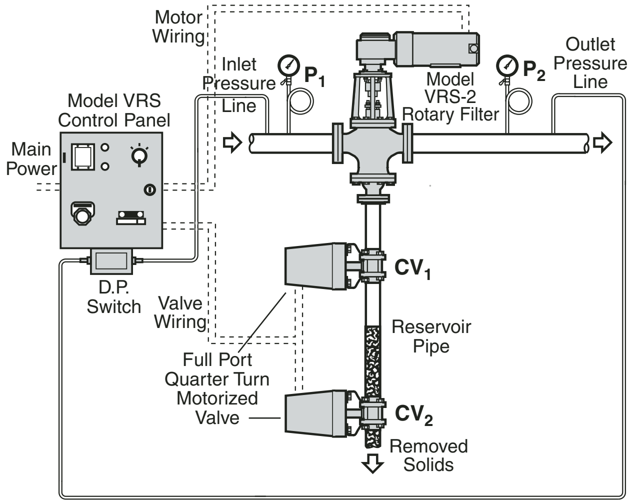


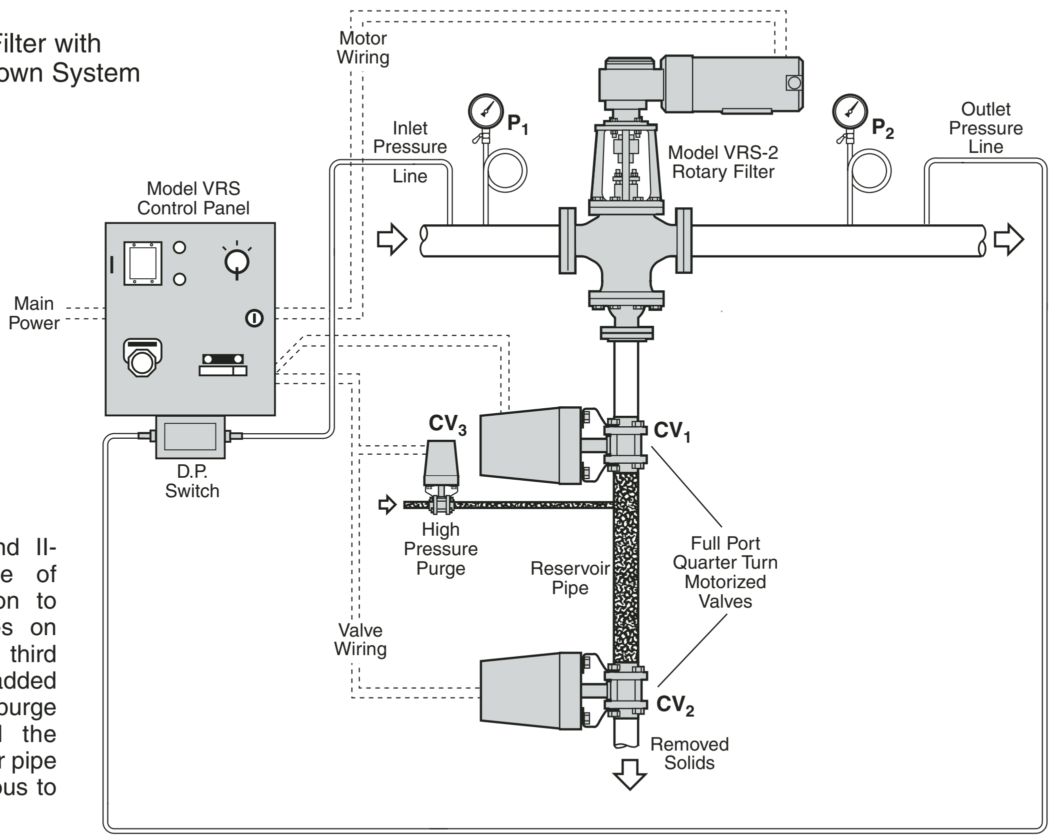
Figure II-100B



The Control Panel operates rotor and blowddown valves CV_1 and CV_2 automatically. Fig. II-100A shows the normal running mode with CV_1 valve open and CV_2 valve closed, to allow reservoir to fill with dirt and debris. As the reservoir pipe fills, valve CV_1 closes and valve CV_2 opens purging only material held in the reservoir leg, Fig. II-100B. CV_2 closes and CV_1 reopens returning system to normal running mode with no stoppage of flow. This hook-up illustrates an automatic filtration system, providing continuous product flow with virtually no loss of usable fluid.

Figure II-101

Motorized Rotary Filter with Three Valve Blowdown System for Viscous FLuids



See Fig. II-100A and II-100B for sequence of operation. In addition to the blowdown valves on the reservoir pipe, a third valve CV_3 has been added for a high pressure purge fluid or air should the debris in the reservoir pipe prove to be too viscous to flow by gravity.

Figure II-102

Freeze Proof Safety Shower
with Antiscalding Protection

Fit T-44 control (with bypass closed), 85˚F to 135˚F range, in 1-1/4" or larger pipe. Flow crosses sensor to shower, to cooling valve inlet ending at the #8 that opens when ambient drops below 40°F. Line flow prevents both freeze up and solar over-heating.

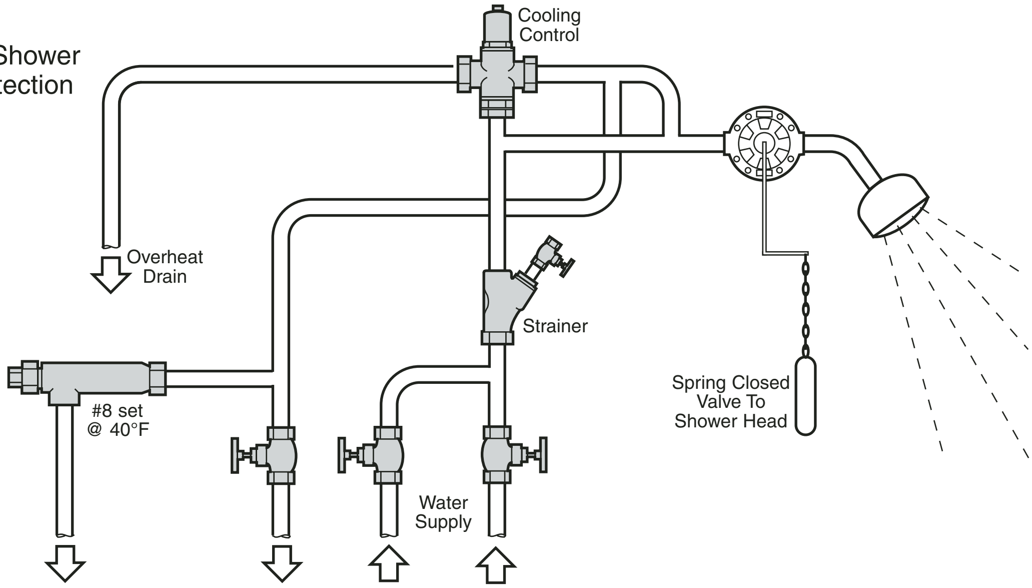
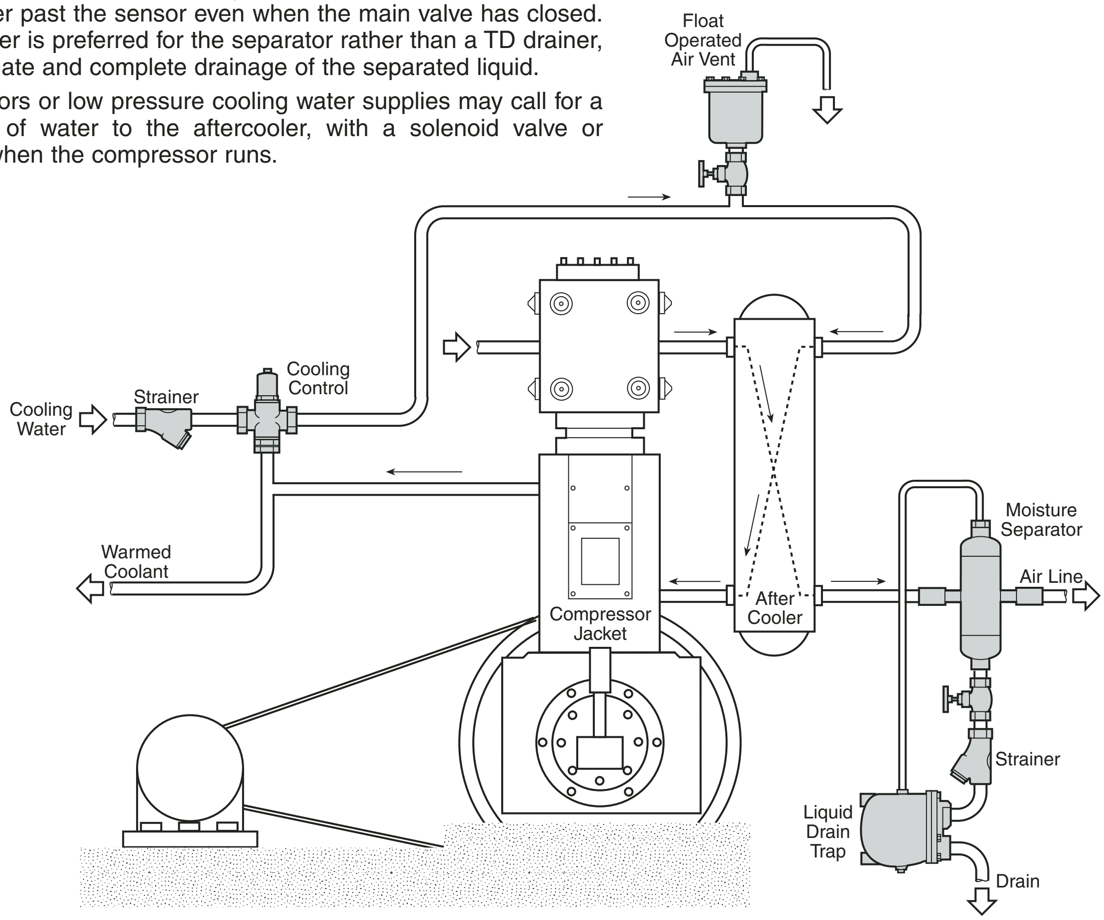


Figure II-103

Automatic Contol of Smaller Compressor
Cooling with Overheat Protection

The T-44 control valve incorporates a bypass needle valve to keep a minimum flow of water past the sensor even when the main valve has closed. A float-type drainer is preferred for the separator rather than a TD drainer, to ensure immediate and complete drainage of the separated liquid.

Larger compressors or low pressure cooling water supplies may call for a separate supply of water to the aftercooler, with a solenoid valve or similar, to open when the compressor runs.



HOOK-UP DIAGRAMS

Figure II-104
Condensate Cooling System

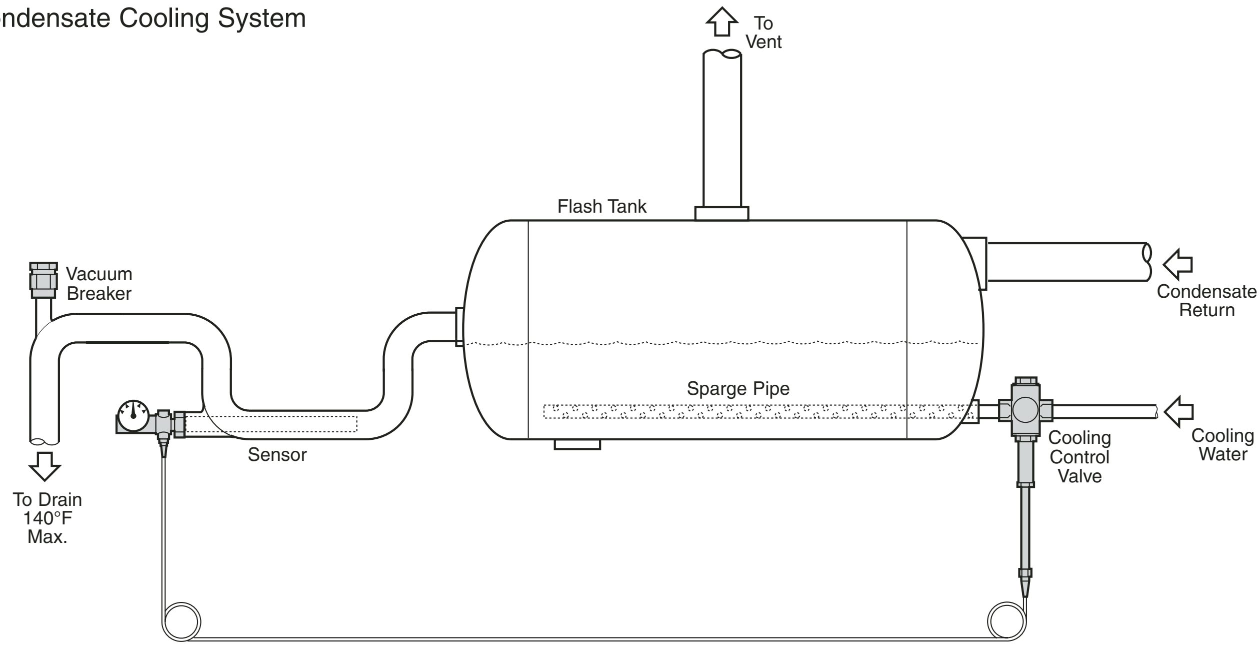


Figure II-105
Condensate Cooling and Flash Knockdown System

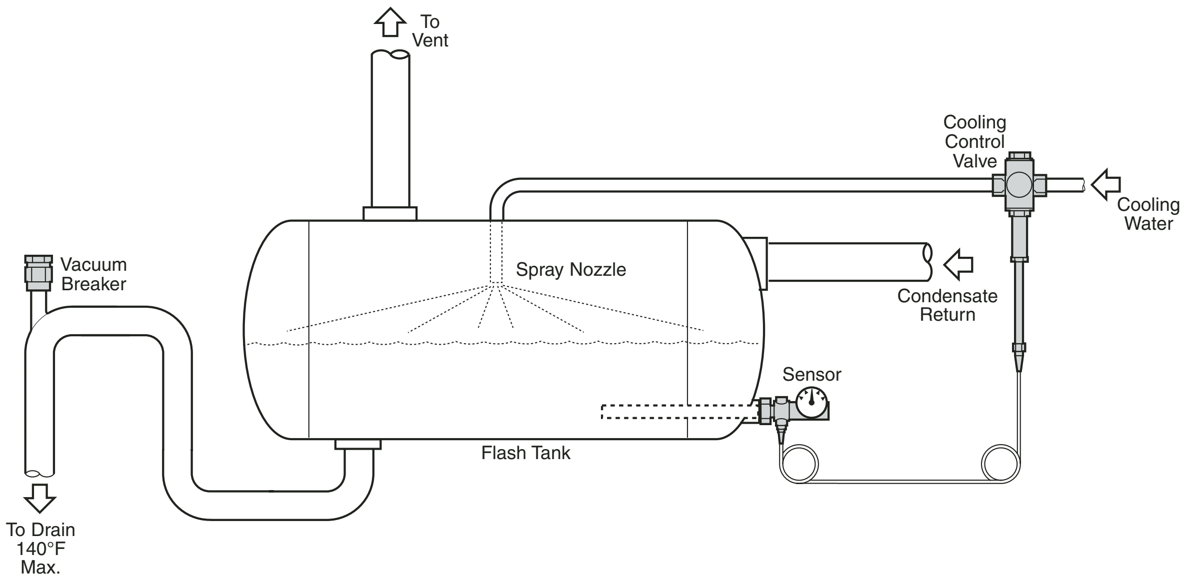


Figure II-106
Continuous Boiler Blowdown
Cooling System

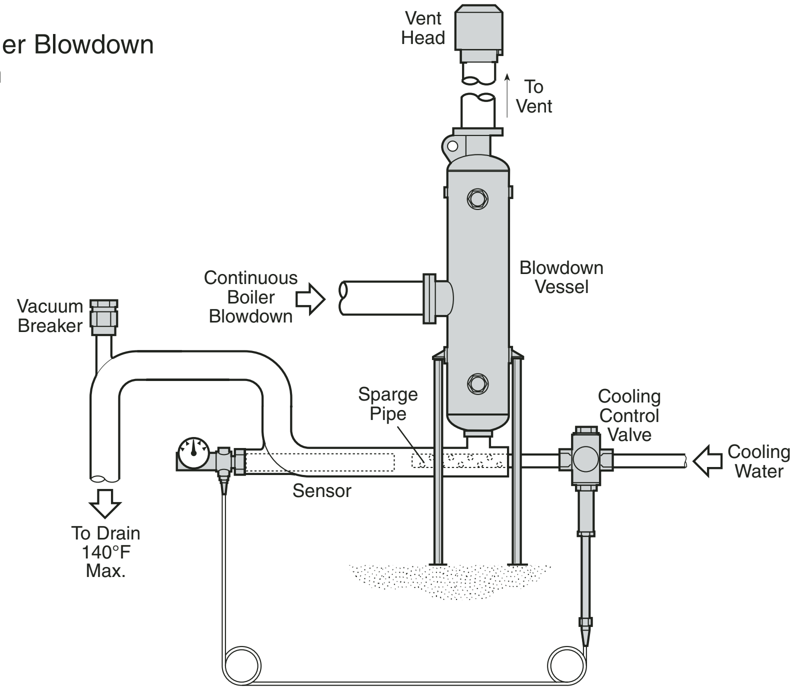
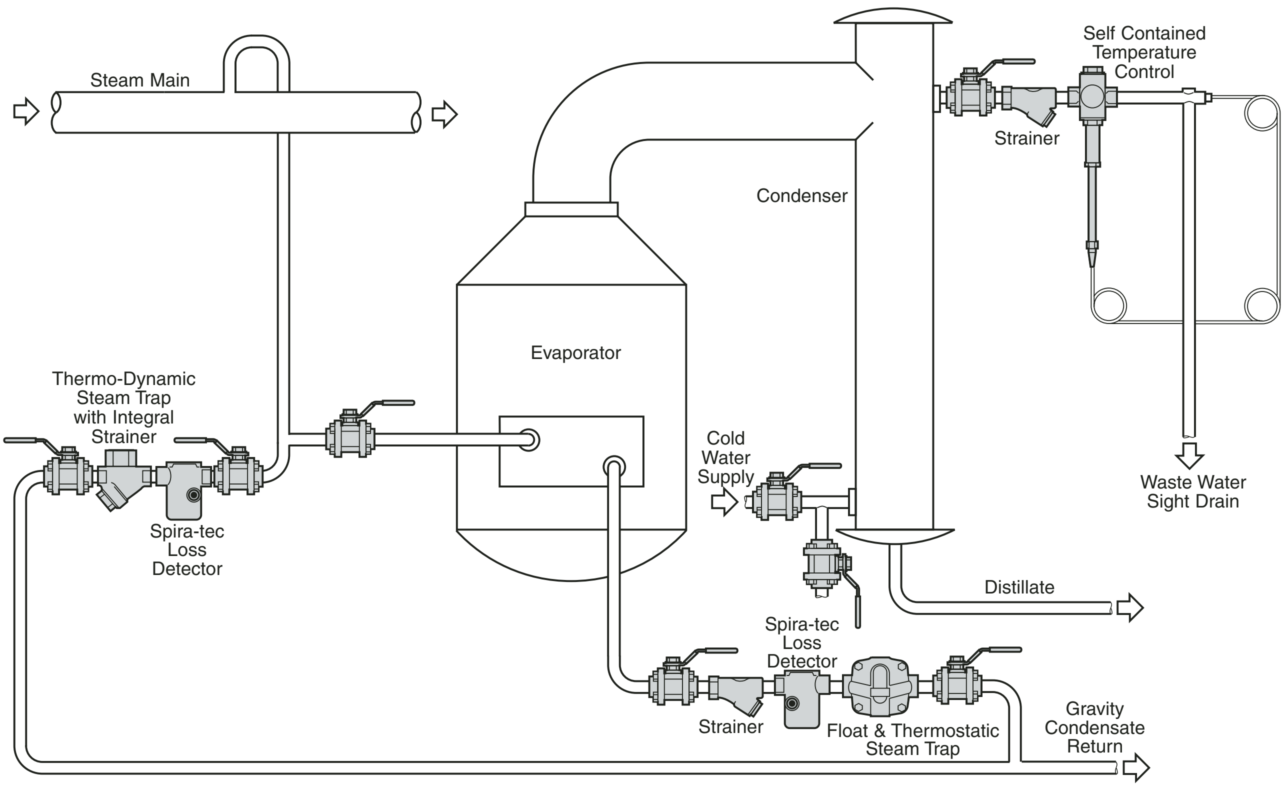


Figure II-107
Controlling Coolant Flow to Vacuum Still
Condenser and Draining Evaporator



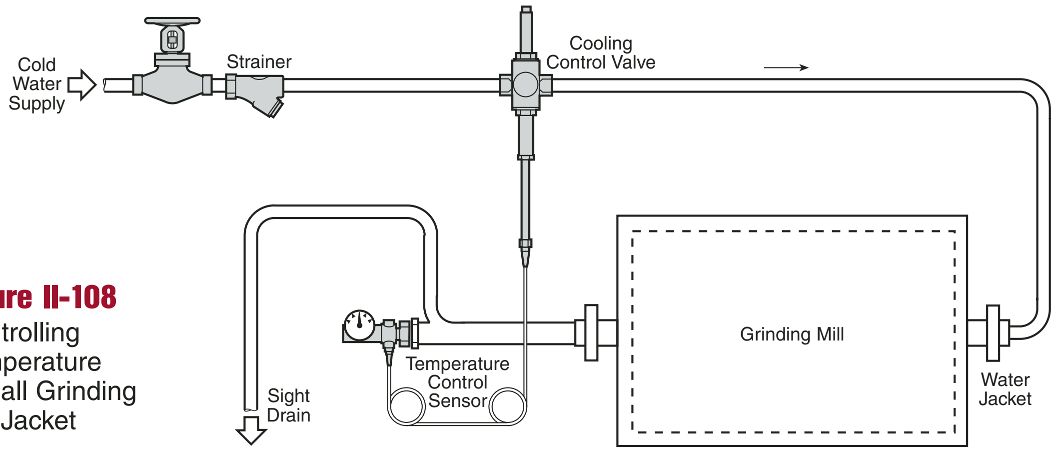


Figure II-108
Controlling Temperature of Ball Grinding Mill Jacket

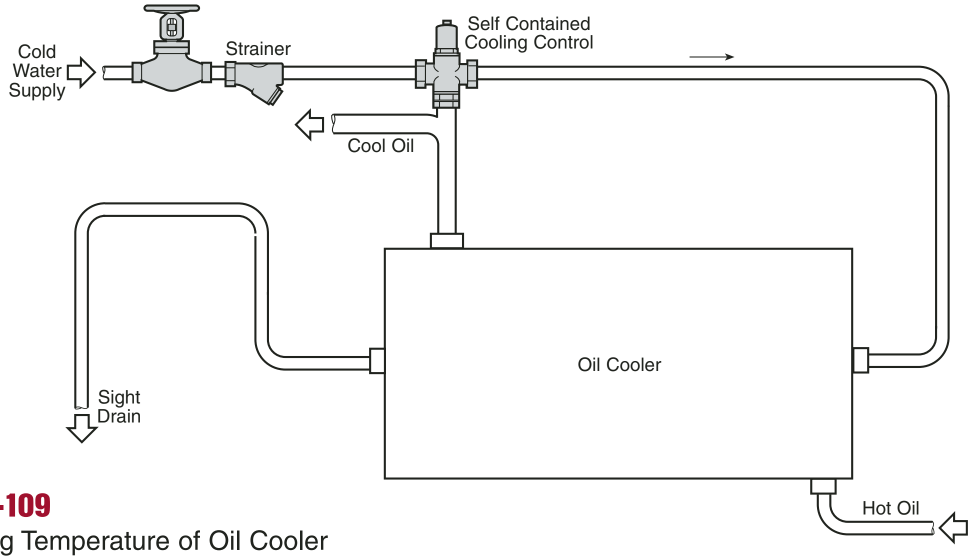


Figure II-109
Controlling Temperature of Oil Cooler

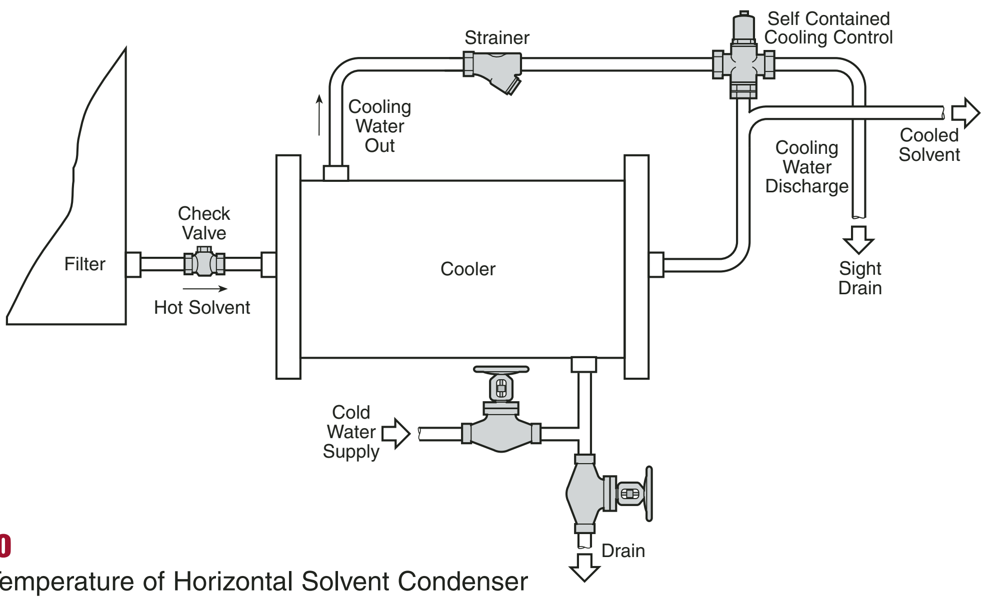


Figure II-110
Controlling Temperature of Horizontal Solvent Condenser

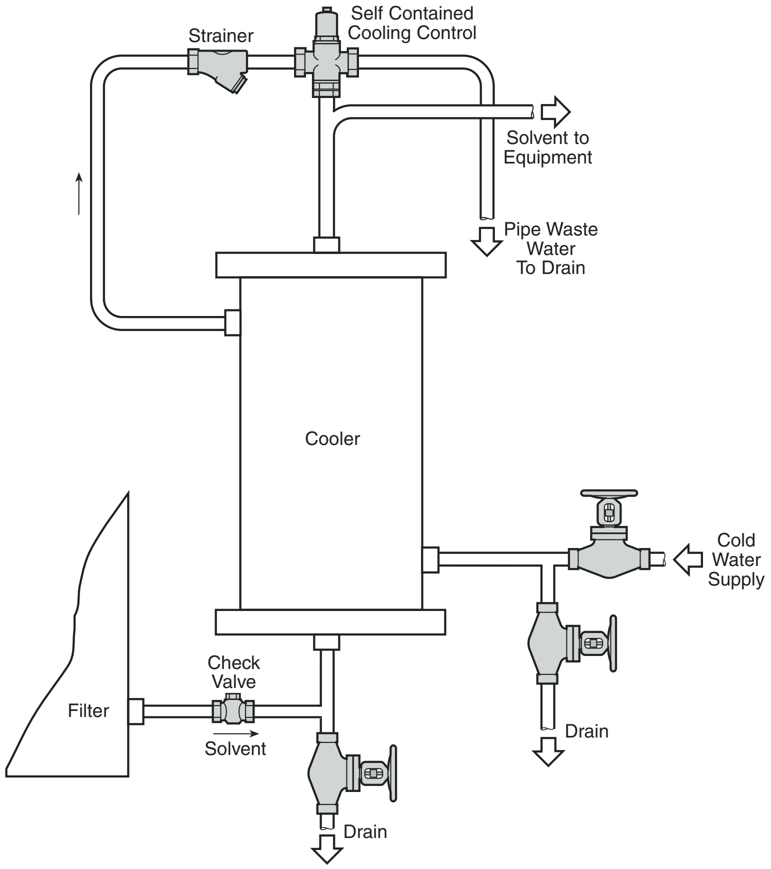


Figure II-111
Controlling Temperature of Vertical Solvent Still

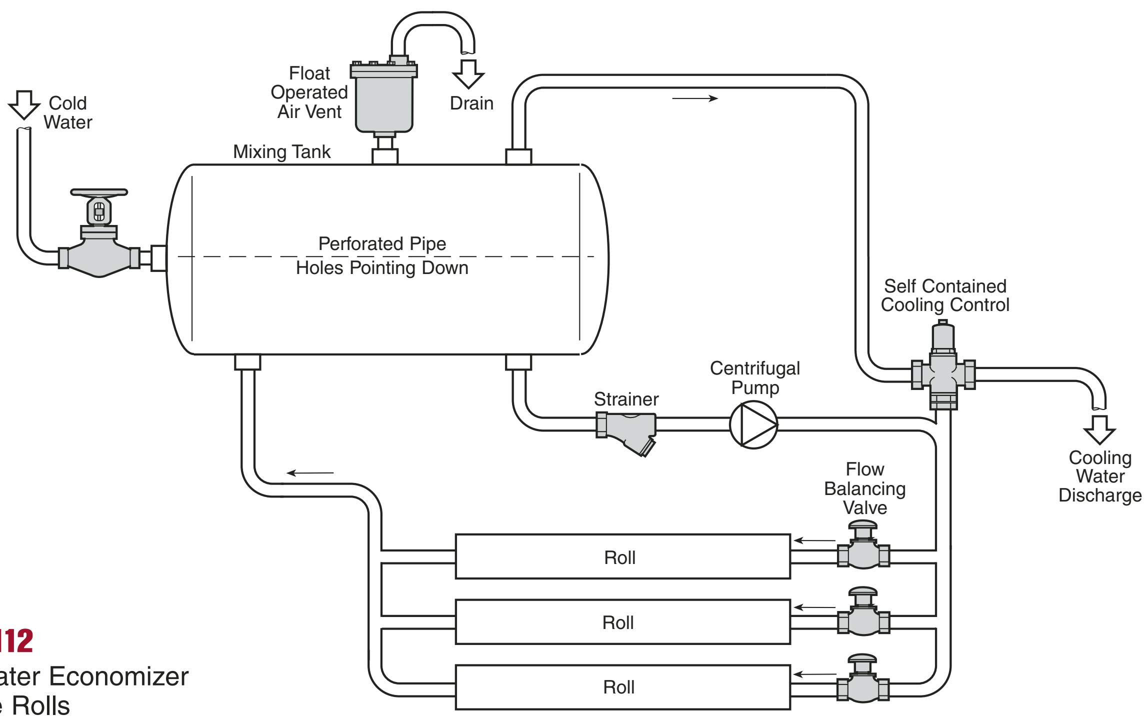
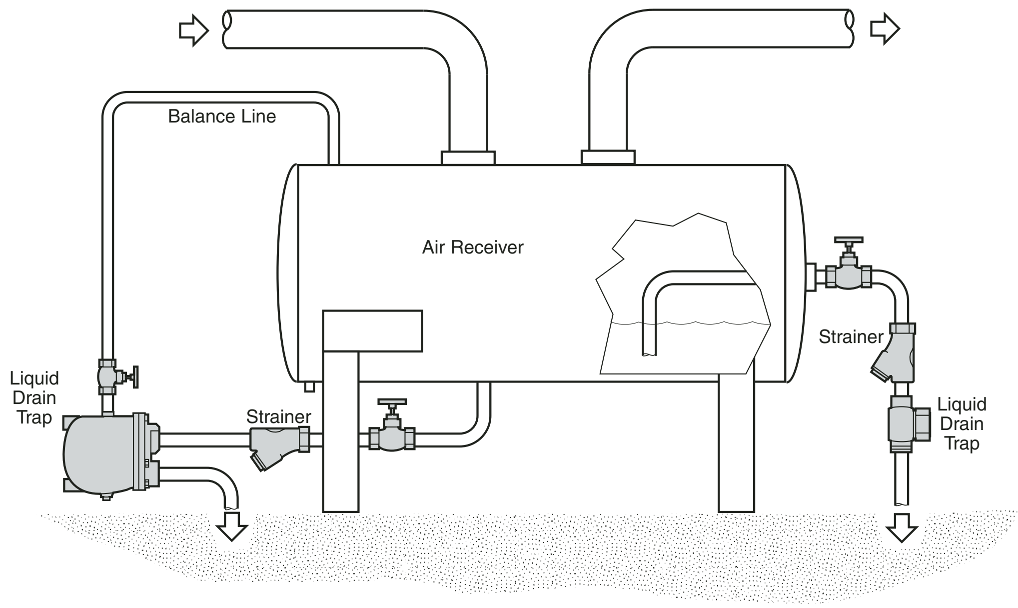


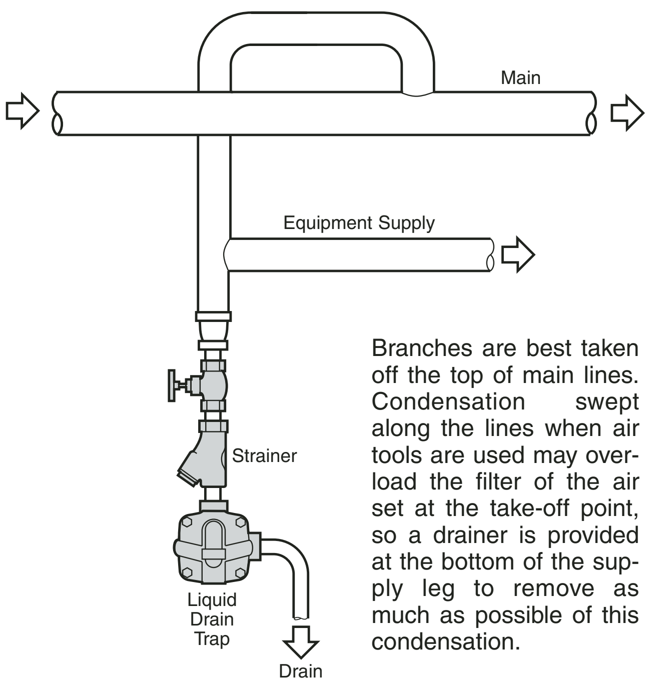
Figure II-112
Cooling Water Economizer for Multiple Rolls

Figure II-113
Alternate Methods of Draining Compressed Air Receiver



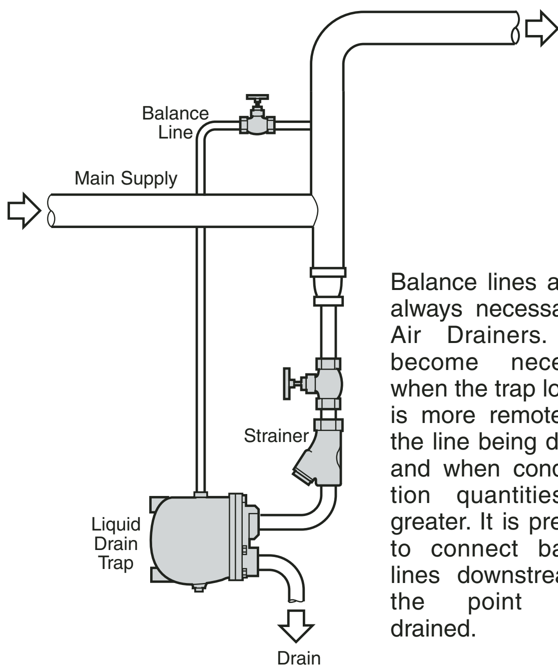
Small air receivers are often “drained” through a manual valve at low level on a once per day basis. Continuous drainage helps to maintain better quality in the air supplied but small receivers may be mounted so low as to preclude the use of the CA14 or FA pattern drainers. The drain point may be in the center of the dished end of even on top, with an internal dip pipe to reach the collected liquid. The only possible option is the TD drainer.

Figure II-114
Draining Compressed Air
Dropleg to Equipment



Branches are best taken off the top of main lines. Condensation swept along the lines when air tools are used may overload the filter of the air set at the take-off point, so a drainer is provided at the bottom of the supply leg to remove as much as possible of this condensation.

Figure II-115
Draining Riser in Compressed
Air Distribution Line



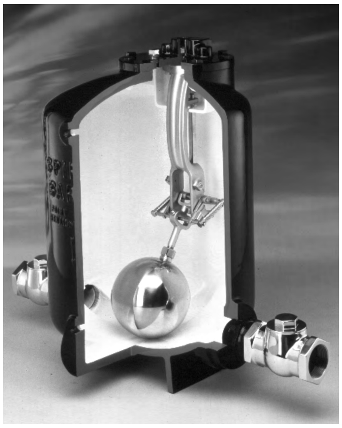
Balance lines are not always necessary on Air Drainers. They become necessary when the trap location is more remote from the line being drained and when condensation quantities are greater. It is preferred to connect balance lines downstream of the point being drained.

PRODUCT INFORMATION

Section 3

spirax sarco®

Condensate Recovery



- Pressure Powered Pumps™
- Packaged Pressure Powered Pumps™
- High Capacity Pressure Powered Pump™
- Electric Pumps

Spirax Sarco offers the solutions to maintaining efficiency in all areas of condensate recovery. For the total system solution, Spirax Sarco's non-electric pumps drain and return condensate and other liquids from vacuum systems, condensers, turbines, or any other steam condensing equipment. The Pressure Powered Pump™ can handle liquids from 0.65 to 1.0 specific gravity and capacities up to 39,000 lb/hr. Available in cast iron or fabricated steel (ASME code stamped) with stainless steel internals and bronze or stainless steel check valves, the rugged body allows a maximum pressure of 125-300 psig and a maximum temperature of 450˚F. This system solution saves energy and provides optimum system efficiency with low maintenance.

For easy installation, Pressure Powered Pump™ packaged units are pre-piped and combine any Pressure Powered Pump™, up to 3" x 2" with a receiver.

Spirax Sarco's electric pumps are packaged units completely assembled, wired and tested. Electric condensate return pumps are available in simplex units with an integral float switch or a mechanical alternator on the duplex units.

Controls & Regulators



- Automatic Control Valves
- Direct Operated Temperature Regulators
- Direct Operated Pressure Regulators
- Pilot Operated Temperature Regulators
- Pilot Operated Pressure Regulators
- Safety Valves

Maximum productivity requires delivering the steam at its most energy efficient pressure and temperature resulting in optimum energy usage, and a safe, comfortable environment. Spirax Sarco has a complete range of controls to efficiently provide the right heat transfer for any process or heating application.

Ranging in sizes from 1/2" to 6", operating pressures up to 600 psi, and capacities up to 100,000 lb/hr, the complete range of controls and regulators provide steam system solutions industry wide. Available in iron, steel, stainless steel, and bronze, Spirax Sarco controls and regulators are suitable for virtually all control applications.

Service Capabilities



- Steam Trap Surveys
- Steam System Audits
- Contracted Site Management
- Model VRS Control Panel
- Steam System Management

When the question is steam the answer is Spirax Sarco. Strategic alliances with Spirax Sarco have benefited many of the world's largest steam users through energy savings, process improvement and out sourcing of non-core activities.

Steam Traps

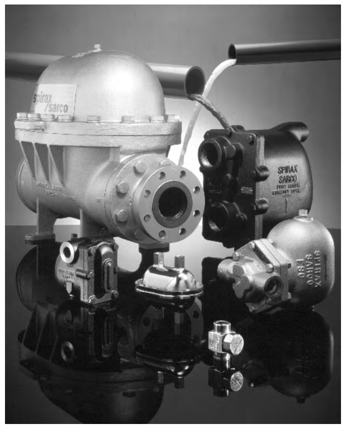


- Balanced Pressure Thermostatic
- Float & Thermostatic
- Thermo-Dynamic®
- Steam Trap Fault Detection Systems
- Bimetallic
- Inverted Bucket
- Liquid Expansion
- Steam Trap Diffuser

Spirax Sarco designs and manufactures all types of steam traps in a variety of materials. Whether it be for steam mains, steam tracing, or heating and processing equipment, Spirax Sarco has the knowledge, service and products to improve your steam system.

Mechanical steam traps are available in iron and steel with NPT, socket weld, or flanged connections in sizes ranging from 1/2" to 4". Thermostatic types are available in brass, forged steel, stainless steel, cast alloy steel with stainless steel internals with NPT and socket weld connections and are available in sizes 1/2" to 1-1/2". The kinetic energy disc types are available in stainless, alloy and forged steel and range in sizes from 1/2" to 1" with NPT, Socket Weld and ANSI connections.

Liquid Drain Traps



Many industrial processes involve the removal of a liquid from a pressurized gas. Spirax Sarco Liquid Drain Traps are ideally suited for this purpose as well as removing condensate from compressed air lines. The float operated design instantly and automatically adjusts to variations in liquid load and pressure.

The traps can handle liquids with a specific gravity as low as 0.5. Liquid Drain Traps have a maximum operating pressure to 465 psi and range in size from 1/4" to 4" with capacities of up to 900,000 lb/hr. Construction is cast iron, ductile iron, carbon steel or 316L stainless steel bodies with NPT, socket weld or flanged connections.

Pipeline Auxiliaries



- Flash Vessels
- Strainers - Pipeline and Basket
- Air Handling Equipment
- Radiator Valves
- Ball Valves
- Steam Injectors
- Steam Separators
- Sight Glasses/Checks
- Trap Diffusers
- Vent Heads
- Vacuum Breakers

The Spirax Sarco line of Pipeline Auxiliaries complete the steam system and are available in a variety of materials and sizes to suit your needs.

Stainless Steel Specialty Products

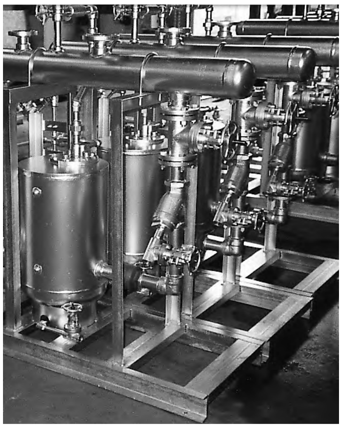


A comprehensive range of stainless steel products:

- Steam Traps
- Separators
- Hygienic Ball Valves
- Pressure Controls
- Filters
- Sample Coolers

The use of clean or pure steam to reduce the risk of product or process contamination spans many industries and applications, including pure steam for sterilization of equipment in the biotechnology and pharmaceutical industries, culinary steam for direct cooking and heating of foods, clean steam for humidification of clean rooms, and filtered steam for hospital sterilizers. Spirax Sarco's range of stainless steel specialty products have been designed and manufactured to the highest standards and specifications required to withstand the rigors of service in clean steam and other aggressive process fluids.

Engineered Systems



Complete modular solutions for steam users worldwide:

- Preassembled Steam Trap Stations
- Steam Distribution and Condensate Collection Manifolds
- Forged Steel Manifolds
- Process Condensate Removal Modules
- Condensate and Flash Steam Recovery Modules

From institutional condensate recovery applications to draining critical process heat transfer equipment, Spirax Sarco's modular pumping systems are the most cost effective and provide the lowest total installed cost. The conventional method of individually specified and procured components with on-site assembly is labor intensive and not conducive to today's competitive plant standards.

The Engineered Systems Advantage expedites the installation process and delivers a quality solution to numerous types of steam users. Each modular pumping system utilizes reliable Pressure Powered Pump™ technology and saves 25% over the conventional method. Spirax Sarco backs each unit with a sole source guarantee and unequaled expertise in steam system technology.

Training



Years of accumulated experience has enabled the development and nurturing of in-depth expertise for the proper control and conditioning of steam. Experienced field personnel work closely with design, operations, and maintenance engineers, continuously evaluating ways to improve productivity. Often, these solutions pay for themselves many times over.

The four U.S. training centers located in Chicago, Houston, Los Angeles, and Blythewood, SC, have on-site steam systems providing hands-on training. Education programs include the theory of steam, the application of steam products, and plant design and system efficiency, to name just a few. Programs also can be tailored to meet individual needs. Thousands of engineers complete Spirax Sarco training programs each year and return to continue broadening their knowledge of steam systems.

Subject Index

Subject Index

Subject Index

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